OFFICIAL PUBLICATION

JOURNAL

Heating • Refrigerating • Air Conditioning • Ventilating

AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR CONDITIONING ENGINEERS

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Word-and-Picture News Report of the Denver Annual Meeting — page 61

How Face Air Velocity Affects Airplane Passenger Comfort — page 41

Heat Flow Problems Can Be Solved by an Hydraulic Analogue — page 45

Counter-flow Air Through Porous Media Provides Effective Thermal Barrier — page 83

Solar Heat Gain Study Projected to Plastics Domed Skylights — page 51

AUGUST 1961

MODEL 237 CRANKCASE
PRESSURE REGULATING VALVE

the best GOOD WILL insurance you can build in...

the surest protection your customers can get

For low-cost overload protection that reflects *quality* in your product and cuts in-warranty service costs, make this valve a *must* in every system you ship.

What happens to your product under conditions of severe overload?

It keeps right on performing if the compressor is protected with an AP Model 237 pressure regulating valve.

There's no chance of motor overload, short-cycling or burnout due to high load conditions. Refrigerant flow is automatically modulated when compressor suction pressures exceed safe levels.

The Model 237 valve installs quickly (in any position) in the system low side between the evaporator and compressor. Adjustment is a one-hand, one-minute job. Available in 1½ and 3 T.R.-12 sizes — ½" or ⅝" SAE flare or ⅝" to 1⅜" ODF; also long connections for silver soldering. Ask your CC representative for Bulletin MI-260 for complete information.

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AUGUST
1961

VOL. 3 NO. 8

OFFICIAL PUBLICATION

ASHRAE JOURNAL

Formerly Refrigerating Engineering including Air Conditioning, and incorporating the ASHAE Journal.

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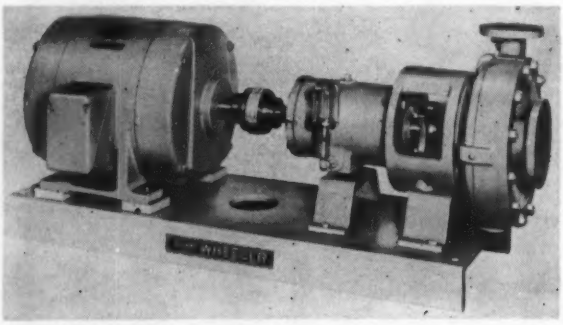
PARTS AND PRODUCTS

CENTRIFUGAL PUMPS

End suction centrifugal pumps, Type BHO, are offered in fourteen sizes, ranging from one to five in. Cast iron alloys and bronzes are used to meet require-



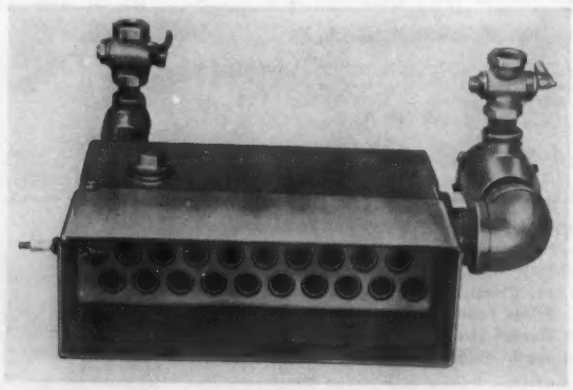
ments for general process applications. Units are capable of pumping liquids ranging in temperature from sub-zero to 350 F at pressures to 250 psi and all can be close-coupled using standard motors with Nema shaft extensions.

Packing or either internal or external mechanical shaft seals can be supplied with the pumps. Five ⅜-in. rings of packing plus seal cage can be accommodated in the deep stuffing boxes. At maximum radial thrust, the shaft deflection at the face of the stuffing box is less than 0.002 in. Two types of frames are utilized: the smaller are equipped with a large four-bolt foot and the larger have extra support provided near the coupling in addition to support near the volute.

C. H. Wheeler Manufacturing Company, 19th St. and Lehigh Ave., Philadelphia 32, Pa.

AIR TUBE BURNERS

Designed for air heating applications having maximum ambient air temperatures up to 750 F, Star



Sixties air tube atmospheric-type burners have capacities (based on 100-F air moving past the burner) ranging from 70,000 to 1,360,000 Btu/hr. Special alloy construction can be provided to withstand temperatures to 900 F. Advantages cited include: simple installation, elimination of all refractory materials, self-contained pilot, excellent turndown characteristics, relatively short flame length at rated capacities and pressure drops and freedom from backfire.

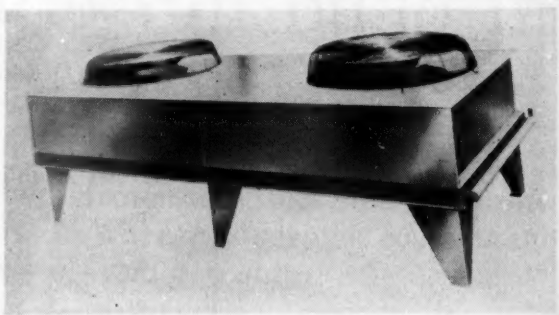
Units are cited as operating equally well on the suction or pressure side of heater fans or blowers. Normal operation utilizes airflow rates of 1500 to 3000 fpm, equivalent to pressure drops of 0.14 to 0.55 in. wc across the burners. Pressure drop and velocity of air passing through and around the burner can be controlled through use of profile plates. Gas volume is controlled by a single valve in the gas line, which may be set manually or automatically.

Eclipse Fuel Engineering Company, Combustion Div, Rockford, Ill.

AIR-COOLED CONDENSERS

Including models for both horizontal or vertical air flow (shown), the BUHV Series is for use in commercial and industrial air conditioning applications. Units are designed for either roof top or ground level installation and are available in ratings from 20 to 60 ton.

Inner-fin-coil construction has permitted compact design of the condensers, by reducing the in-

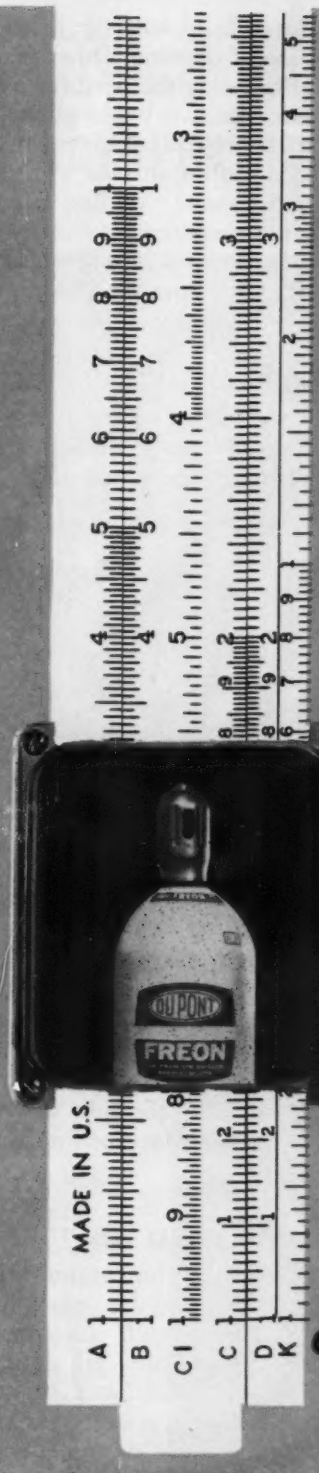


ternal volume of the condenser and requiring less refrigerant charge. Low static pressure drop is cited as being assured with less fan hp and lower operating costs. Aluminum fins are mechanically bonded to copper tubes in the units and copper headers are used throughout. Slow-speed, belt-driven propeller fans are of the non-overloading type. Motor is drip-proof and grease-lubricated. Fan sheave, motor sheave and V-belts are supplied and mounted with each unit.

Westinghouse Electric Corporation, Air Conditioning Div, Staunton, Va.

MINIATURE SOLENOID VALVE

Developed for hydraulic and pneumatic service, this brass, miniature, three-way solenoid valve, designated BM-300, can handle differential pressures from zero to 200 psi or up to 500 psi for special applications. It is available in normally-closed, normally-open, directional and free-venting versions. A new design that improves the flux path and heat dissipation is cited as having made it possible to reduce the overall size of



How can you measure the quality of a refrigerant?

Measure refrigerant quality by performance and time. When both are considered, they narrow the field to but one choice—Du Pont "Freon".

In actual performance **"Freon", the first fluorocarbon refrigerant, has been proven in millions of air-conditioning and refrigeration systems—has established a standard of refrigerant performance that has never been surpassed.**

In actual time **"Freon" has been proven by 30 years of increasing use in all types of air-conditioning and refrigeration equipment—years during which no other type of refrigerant has been able to challenge Du Pont's original fluorocarbon. Today's air-conditioning and refrigeration industry is based on the development of "Freon".**

In addition, "Freon" is backed by a continuing program of technical and marketing assistance designed to improve the performance of air-conditioning and refrigeration systems and expand the future market for these systems.

Any way you measure, it pays to use Du Pont "Freon". Specify it as the refrigerant in your equipment. It will never let you or your equipment down.

FREON® ***premium quality refrigerants***

DU PONT

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the valve while maintaining the same flow capacity and working pressure ranges of large three-way solenoid valves.

Measuring 1-3/16 in. diam, the cylindrical body is ⅞ in. deep. Five body orifice diam of 3/64, 1/16, 3/32, ⅛ and 5/32 in. have C_v factors of 0.05, 0.12, 0.21, 0.32 and 0.40, respectively. Upper orifices with 1/16 and 3/32 in. diam have corresponding C_v factors of 0.08 and 0.12. Standard valves handle liquids and gases at temperatures from –45 to 150 F, with special models available for media temperatures of –60 to 350 F. Coil voltages are available from 6 to 440 volt ac (50 to 60 cps) or 6 to 220 volt dc. Standard coils are rated to 105 C and a maximum ambient temperature of 25 C for continuous duty; special coils for service to 220 C are rated to 115 C for continuous duty.

Allied Control Company, Inc., 2 East End Ave., New York 16, N. Y.

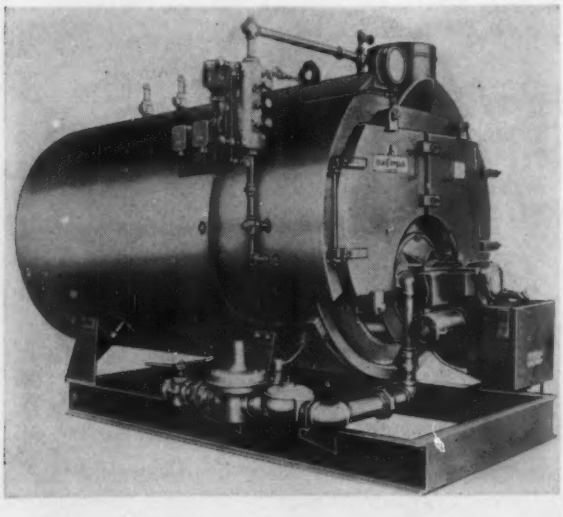
SLIME CONTROL

Odorless and non-toxic, Vaporene R is cited as killing slime growths in air conditioning systems, and as being suitable also for use in hospitals and food processing plants. Offered with the product is a test kit to enable the user to tell when the correct amount of chemical has been added to the water in a system.

Metropolitan Refining Company, Inc., 50-23 23rd St., Long Island City 1, N. Y.

FOUR-PASS GENERATOR

Two types of this package unit are offered: the 400L (fired by gas, oil or gas-oil) is a low pressure model with a rated output of 50 through 100 hp and is designed for 15-psi steam or 30-psi water; the 400H



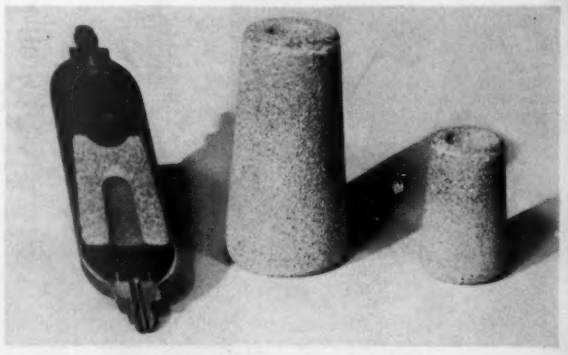
(also gas, oil or gas-oil) is a high pressure model designed for 100 through 150-psig steam, with a rated output of 30 through 100 hp. Elimination of refractory baffles is achieved by wet back design. Provision of individual heads for each tube pass is cited as minimizing tube sheet stress caused by uneven heat distribution.

Fireside and waterside surfaces are easily accessible to allow minimum maintenance. Front access doors, lined with two in. of refractory, are hinged for ease of opening. Other features include: forced draft eliminating the need for a high stack or induced draft fan; positive water circulation through internal distributors and arrangement of tubes; five sq ft of fireside heating surface with each hp; all tube ends rolled and beaded, welding is not required; and low electrical requirement.

Iron Fireman Manufacturing Company, 3299 W. 106th St., Cleveland 11, Ohio.

REFRIGERATION CARTRIDGES

Used in Filter-Kore drier cartridges is Molecular Sieve Type 4A in the form of solid cores, which act as mechanical filters as well as desiccants. Because of their



ability to remove traces of acid from the refrigerant stream in addition to water, they offer protection to the system from acid in the event of motor burnout or refrigerant decomposition. Filtration is provided down to a 50-micron particle size. Pressure drop through a ½-in. section of core is under two psi, as measured with trichloethylene at a flow rate of five psi/min. Density of the fully activated cores is 55 lb/cu ft.

Kenmore Machine Products, Inc., Lyons, N. Y.

INFRA-RED HEATERS

Developed for farm utility heating applications, these units use natural or propane gas for fuel and are cited as having high efficiency in high intensity infra-red radiation. Infra-red, since it warms objects directly on absorption rather than transferring heat by warming the air first, is suited for heating farm buildings, where frequent air movement increases operating costs by carrying away warm air.

Hupp Corporation, Fannon Products Div, Detroit, Michigan.

REGISTERS, GRILLES, DIFFUSERS

Featuring extruded, anodized aluminum frames combined with all-nylon moving parts, this line of registers, grilles and curved blade diffusers is available in commercial or residential stock sizes. Non-combustible, the unbreakable nylon has a heat resistance of more than 300 F and, because of its sound barrier characteristics, permits higher jet velocities without whistling or roaring. Additionally, it is virtually condensation free.

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ALCO EVAPORATOR PRESSURE REGULATORS

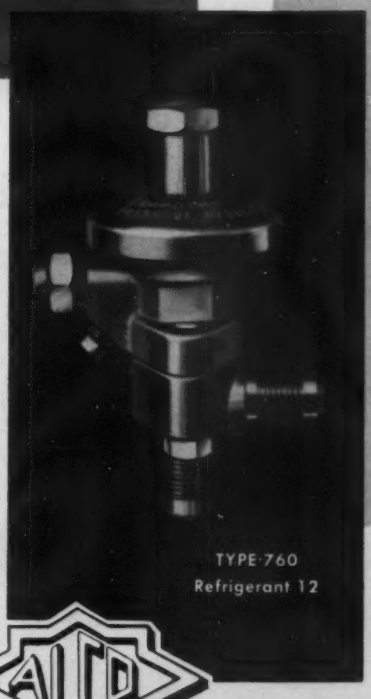
TYPES 15, 16, 18
Refrigerant 12
Refrigerant 22
Ammonia

Your protection from freeze-up

- **SODA FOUNTAINS**
- **BEVERAGE COOLERS**
- **PRODUCE CASES AND PRODUCTS**
- **PLATING BATHS**
- **WATER CHILLERS**
  — from the smallest to the largest

Eliminates tube bursting

TYPE 435
Refrigerant 12



TYPE 760
Refrigerant 12

ALCO'S EVAPORATOR PRESSURE REGULATORS are designed, engineered and manufactured by ALCO, under ALCO'S HIGH QUALITY CONTROL SYSTEM—to efficiently maintain *evaporator pressure* in either a single or multipal system—*regardless of load changes.*

ALCO EPR VALVES are available from ½" to 6" port sizes and all connections up to and including 6⅛" O.D.F.—FOR ALL REFRIGERANTS.

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The one complete line of refrigerant controls: Thermostatic Expansion Valves . Refrigerant Distributors Solenoid Valves . Suction Line Regulators . Flooded Evaporator Controls and Reversing Valves

Frame sections are joined mechanically at the miter with the use of a Corner-Loc device. Gasket sealing is accomplished by using a vinyl gasket extrusion fitted into a special channel in the undercarriage of the frame.

Air Guide Corporation, 275 N. W. 27th St., Miami 37, Florida.

HEAVY DUTY AIR CONDITIONERS

Remote air-cooled condensing units in 90,000 and 118,000-Btu/hr capacities have been added to this manufacturer's line. Designed primarily for commercial and industrial rooftop installations, this heavy-duty equipment has a low silhouette design, with an overall height of 51 in. Condenser coil is mounted horizontally to decrease the height and air discharge

is upward, from a large diameter, belt-driven fan. Choice of fan coil units or horizontal air-flow coils is offered. Accessory motor and drive kits provide an extensive range of air delivery.

Day & Night Manufacturing Company, P. O. Box 2222, La Puente, Calif.

ELECTRIC WATER HEATER

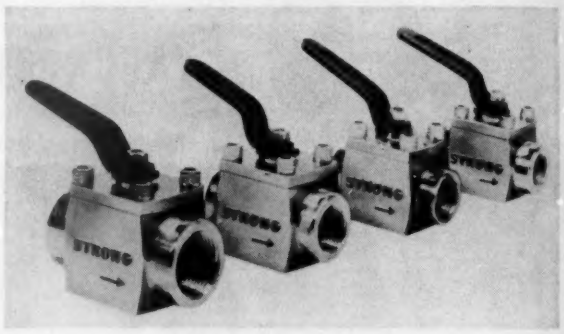
Designed to supply large volumes of hot water, the Marathon features fast recovery and is constructed of ¼-in. steel. Produced in a galvanized or glass-lined heavy duty tank, the unit is available in up to 12,000 watt. Standard voltage is 236, with 208 volt provided on request.

W. L. Jackson Manufacturing Company, Inc., P. O. Box 190, Chattanooga, Tenn.

TOP-ENTRY BALL VALVES

One-piece body design constructed from cadmium-plated carbon steel or stainless steel bar stock is featured on this line for safety, strength and economy. For on-off valving of gas, air, oil, water or liquid chemicals, the units have threaded inlet and outlet connectors for ⅜, ½, ¾ and 1-in. lines. An arrow indicating flow direction is cast into the valve body.

Maximum pressure-temperature ratings for the valves are 150 psi at 365 F for steam lines and 1000 psi for water, oil, liquid chemical and gas handling



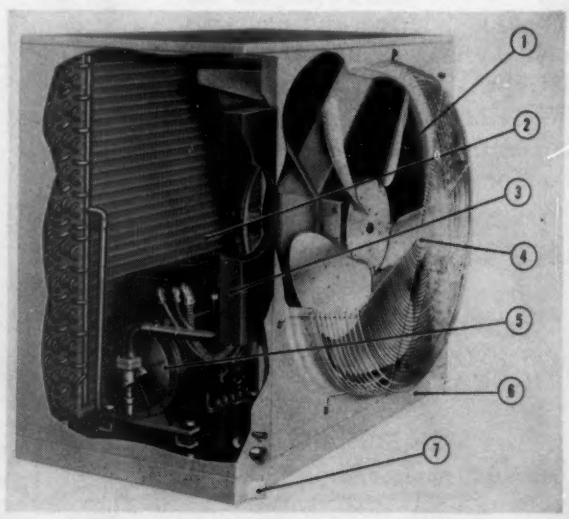
applications. Balls are of hard chrome-plated bronze or stainless steel, resting between seals of Buna N or Teflon. A flow passage is drilled through the ball in diam of 0.500 on the ⅜ and ½-in. sizes, 0.625 on the ¾-in. and 0.824 on the one-in. models.

White Sewing Machine Corporation, Strong Div, 508 Sandusky St., Conneaut, Ohio.

11-TON CONDENSING UNIT

Large condenser air volume and special coil circuiting are utilized by this 11-ton unit to gain maximum capacity. Provided by the coil circuiting is a sub-cooling benefit that delivers liquid refrigerant to the evaporator up to 20 F cooler than the evaporating temperature.

Resiliently mounted, the compressor is statically and dynamically balanced and has light-weight aluminum pistons and rods. Shown in the illustration are: 1, receiver—small diameter vertical design with purge valve and two liquid level valves; 2, condenser — four rows of ½-in. coils and ripple-edged aluminum fins flat-bonded to tubing; 3, control box — readily



accessible inside end panel; 4, axial flow fan — belt-driven, five-bladed, 36-in. unit delivering 8800 cfm and having a three-phase, one-hp motor; 5, semi-hermetic compressor — 220/440 volt, equipped with four-position suction shut-off and two-position discharge shut-off valves; 6, cabinet — main panels are

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PEAK PERFORMANCE PRODUCTS FOR

AMMONIA REFRIGERATION SYSTEMS

THERMOSTATIC EXPANSION VALVES and REFRIGERANT DISTRIBUTORS
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LEVEL MASTER CONTROLS
for Flooded Systems

SOLENOID VALVES

Be sure to get
SPORLAN CATALOG 59A
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It gives you all the facts on
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Ammonia Products

CATCH-ALLS

STRAINERS

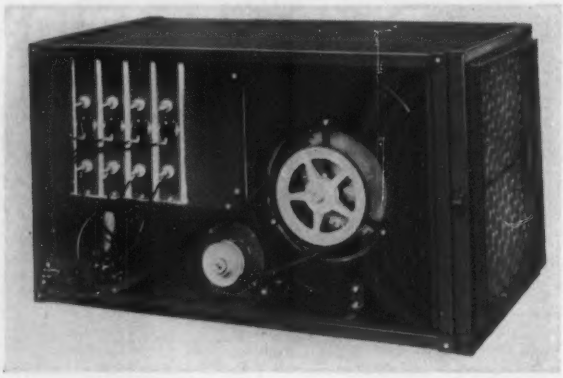
BACK PRESSURE REGULATORS

SPORLAN VALVE COMPANY
7525 SUSSEX AVENUE · ST. LOUIS 17, MISSOURI
EXPORT DEPT. 85 BROAD ST. · NEW YORK 4, N. Y.

16 gauge, corners are 14 gauge with 10 gauge base rails and cross-members; and 7, crating lugs – hold unit firmly in crate for shipping, but are designed also for hoisting unit into place and for bolt-down in permanent position.
Lennox Industries, Inc., 200 S. 12th Ave., Marshalltown, Iowa.

ELECTRIC WINTER CONDITIONERS

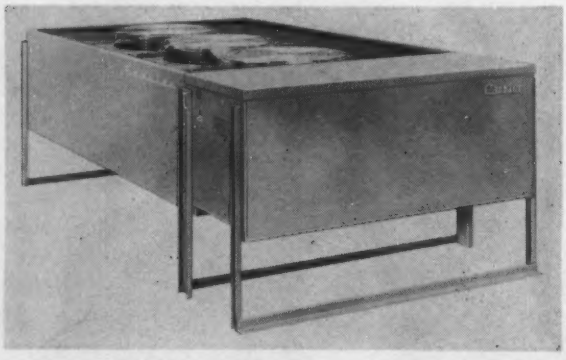
Six units with electrical inputs of from 5 to 35 kw and output ratings ranging from 34,150 to 119,525



Btu/hr comprise the new Series E line of Luxaire Electric Winter Air Conditioning Units. Equipment can be installed in either horizontal (shown with panel removed, revealing wiring terminals, sequencer, control panel, ends of heating elements, blower and motor and slide-out filter frame), upflow or counterflow positions. Cabinets are compact, measuring 34 x 20 x 20 in. for the three smaller sizes or 42 x 25 x 25 in. for the three larger sizes. Smaller capacity units in the series have a ten-in. standard equipment blower and provide air handling capacity for up to four ton of cooling. Larger models are equipped with a twelve-in. blower and have air handling capacities to five ton.
C. A. Olsen Manufacturing Company, Elyria, Ohio.

15-TON HEAT PUMP

Indoor and outdoor (shown) heat pump sections with a nominal capacity of 15 ton have been designed to meet a wide range of climatic conditions and applica-

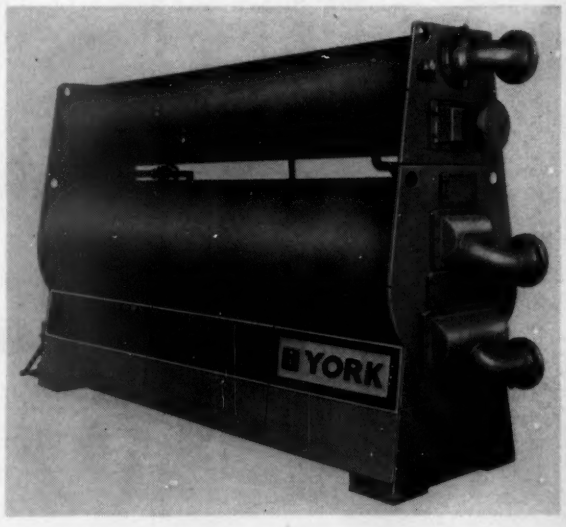


tions. Featured on the outdoor section is a low silhouette, but 20¾ in. high. Three vertical discharge propeller type fans move 13,000 cfm at a low sound level. At 95 F outdoor temperature and 67 F wet bulb indoors, the heat pump produces 173,000 Btu/hr of cooling. At 40 F outdoor and 70 F indoors, it produces 162,000 Btu/hr of heating. As many as three accessory electric heaters of 19.1 kw each may be mounted on the fan deck.

Operating costs are cited as being reduced by a defrost cycle that occurs every hour when outdoor temperatures require it. Rapid cycling is prevented by a Time Guard control circuit. Refrigerant subcooling helps reduce operating cost and enables the outdoor unit to be located as much as 60 ft above or below the indoor unit. Indoor section can be used with or without ductwork and has a matching plenum for free discharge of air into the conditioned space.
Carrier Corporation, Carrier Air Conditioning Company Div, Syracuse 1, N. Y.

WATER CHILLING ABSORPTION UNITS

Offered by these 1961 systems are a number of improvements over previous lines. Model EK, shown, comes in twenty capacities, ranging from 50 to 1000 ton of cooling for air conditioning and process applications. Among features of the line are extensive flexibility of application, low installation cost, im-



proved operating economy, minimum maintenance and excellent safety and quietness.
York Corporation, a Subsidiary of Borg-Warner Corporation, York, Pa.

COMPACT HEATING ELEMENT

Introduced for commercial use as a low temperature warmer or in spot heating applications, a compact glass heating element features a glass panel selectively coated with a metallic film. The film is applied to form an electrically-conductive path, which permits a maximum amount of heat to be generated from a minimally-sized panel. Temperatures in excess of 300 C can be produced with glass sizes from 1 to 27⅜ in. square and from 3/32 to ¼ in. thick. Resistance per square of coating can be varied from 6 to 100 ohm.

Low Water Cut-offs on Hot Water Boilers?!

...the industry has given the answer

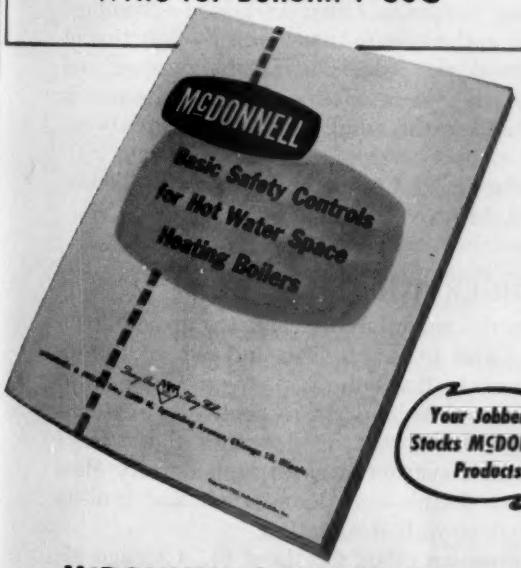
Yes, the answer has come from the field . . . from the engineers, contractors and manufacturers. They have seen the logic of installing a water level control on hot water space heating boilers . . . a low water cut-off, or—for even greater precaution—a feeder cut-off combination.

We knew it was a good idea. Expected it to grow and grow. But frankly we have been surprised at how rapidly the heating industry has taken hold of water level control as a logical team mate for an ASME pressure relief valve.

In fact, many local codes now require a low water cut-off, or feeder cut-off combination, on hot water boilers installed in places of public occupancy—including multiple dwelling units.

Notice the diagrams of recommended installations opposite. For more detailed discussion get this booklet that tells the whole story: "Basic Safety Controls for Hot Water Space Heating Boilers."

The What-Why-How in eight interesting pages
Write for Bulletin P-30C



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McDONNELL & MILLER, Inc.
3500 N. Spaulding Ave., Chicago 18, Ill.

Doing One Thing Well

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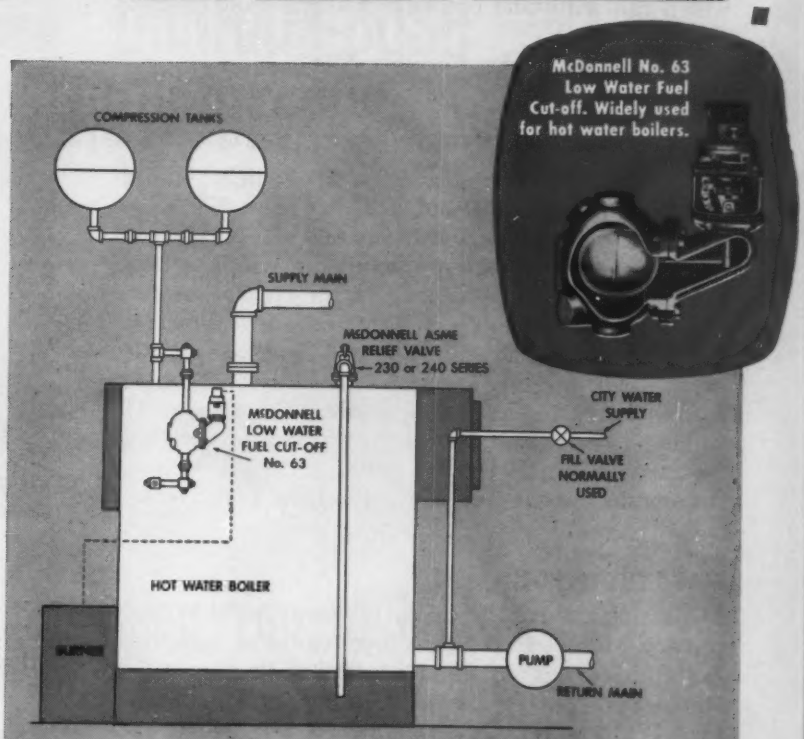
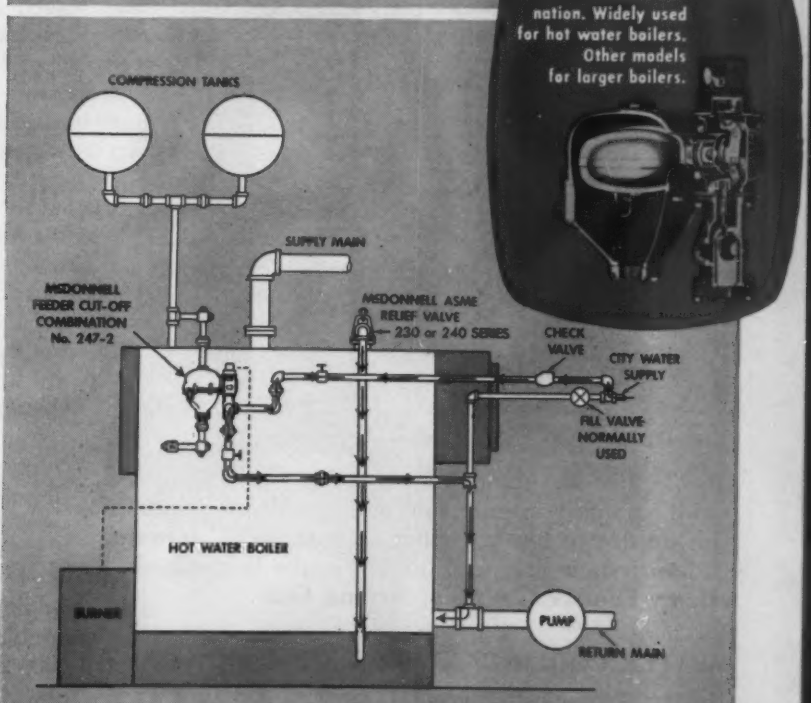
Water Level Controls



1 The essential low water control for a hot water boiler. A McDonnell Low Water Cut-off located above lowest permissable water level; also McDonnell A.S.M.E. Pressure Relief Valve.



2 The safest and most complete control for a hot water boiler. Offers a combination of mechanical and electrical safeguards which is today's best and most complete answer. A McDonnell combination Boiler Water Feeder and Low Water Cut-off; also McDonnell A.S.M.E. Pressure Relief Valve.

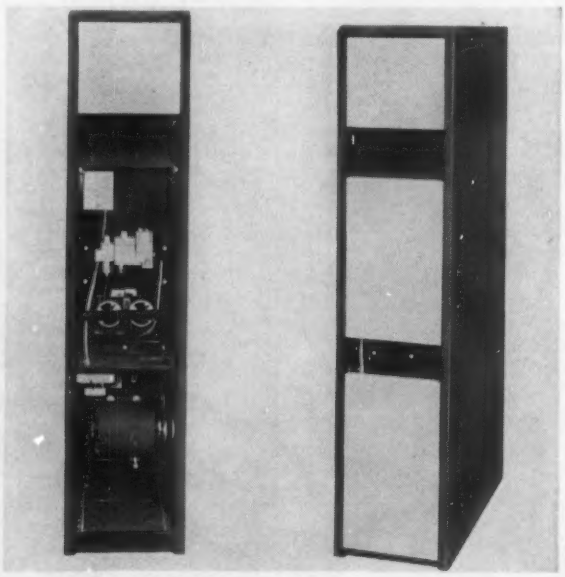
Maximum dissipation is 10 watt per square of coated area. Glass used in the unit is Pyrex 7740, a borosilicate glass which has good insulating properties and which can withstand corrosive materials and thermal shock.
Corning Glass Works, Corning, N. Y.

AUTOMATIC TRANSFER SWITCHES

Mechanically held automatic transfer switches are offered now in 1200, 1600 and 2000-amp sizes. These two and three-pole units are listed to 600 volt ac, 250 volt dc. Switches are open-type, minimum width, vertical tandem types, suited especially for control centers. Full relay protection on each provides for transfer to the emergency source when voltage on any phase drops to 70% or less. Retransfer to the normal will be made when all phase voltages are 90% or more. On single-phase or dc, three-wire systems, normal voltage is supervised across the live lines.
Automatic Switch Company, Florham Park, N. J.

GAS-FIRED FURNACE

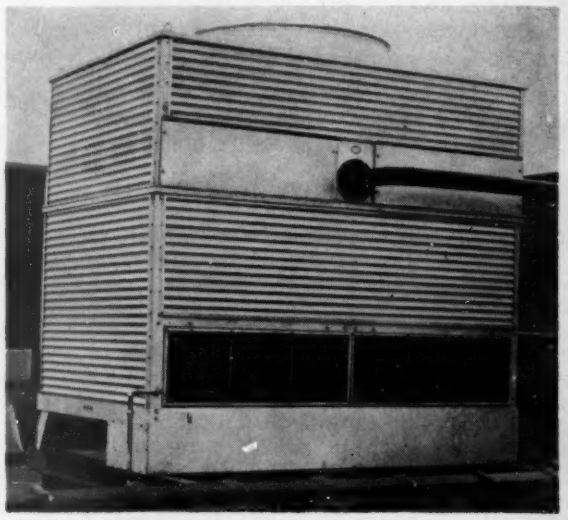
Measuring but 12 in. wide, 26 in. deep and 54 in. high, the new Moncrief SH-50 gas-fired winter air conditioning furnace has an input of 50,000 Btu/hr and is shipped completely assembled and wired at the factory. Featured is a heavily constructed, 16-gauge, end-welded steel heat exchanger. Either a belt-driven or direct-drive blower is available; the standard equipment, nine-in. belt-driven blower, when equipped with a properly sized motor, will provide air deliveries for up to two ton of cooling against 0.5 in. of water external static pressure. Bottom intake is standard.



Henry Furnace Company, Medina, Ohio.

WATER COOLING TOWERS

Covering a capacity range upward from 100 ton, nominal, the CompacTower line is light in weight, utilizing curtain-wall casing, honeycomb segments in filling and a new design of cold water basin to minimize weight. All units are modular, with uniform height and width dimensions permitting easy multi-cell installations. All internal areas are accessible for inspection and maintenance through conveniently located access panels.



Marley Company, 222 W. Gregory Blvd., Kansas City 14, Mo.

MOISTURE ELIMINATING VALVE

Designated Moisture Minder, a new valve that automatically and continuously eliminates water and oil condensation from compressed air lines without causing pressure drop can be attached to existing systems without cutting the lines. Cited as taking advantage of the pressure in the system to perform the function of moisture removal, the valve will not balance off and bleed air from the system. Six in. in length, the unit is 1¼ in. diam and is threaded to fit a standard ¼-in. pipe.
Barger Manufacturing Company, 204 Foshay Tower, Minneapolis 2, Minn.

FOUR REFRIGERATORS

Introduced by this manufacturer are two upside-down models in 12.5 and 14.5-cu ft sizes and two right-side-up refrigerators, in the same capacities. Cabinets of the units are 28 and 32 in. wide, respectively. Featured are flush-hinge doors that open within the cabinet width, thin-wall construction with high density glass fiber insulation, double-seal door gasket and built-in design for flush-to-wall installation.
Admiral Corporation, 3800 Cortland St., Chicago 47, Illinois.

MODULAR DUCT FURNACES

Addition of four new capacities to the DS Series doubles the number of sizes available. Rated at 175,000, 225,000, 275,000 and 350,000-Btu/hr input, the new furnaces supplement the previously available 150,000, 200,000, 250,000 and 300,000-Btu/hr models. Standard equipment consists of condensation shields for burner protection, hooded pilots and drain pans

Worth Remembering When an Air Cooled Condenser is Required...

DUNHAM-BUSH NOW OFFERS 15 SIZES...

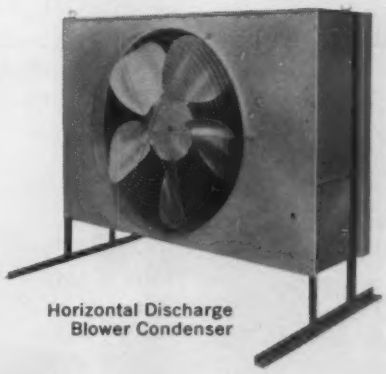
Enabling You to Match the Unit to the Job

No need any longer to specify a larger capacity unit than you need. The extended Dunham-Bush line of Remote Blower Condensers—in 15 horizontal or 15 vertical air discharge models—covers a full nominal capacity range from 5 through 85 tons.

Because its new design ensures low pounds-per-square-foot-loading, the 'LSBC' Low Silhouette Blower Condenser can be installed virtually anywhere on any roof, without the costly reinforcement of roof members usually necessary to support heavier units. And the low, compact 'LSBC' is easily accommodated within minimum space, yet has the operational and construction features to meet the most rigid performance demands. Air movement is up and out, making possible a minimum noise rating.

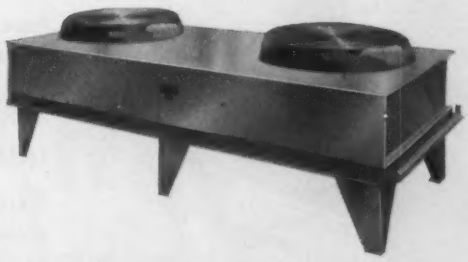
Other major features of both horizontal and vertical discharge units include a high efficiency Inner-Fin condenser coil with less internal volume and therefore smaller refrigerant charge . . . aluminum casing . . . and weather protected motor specifically designed for outdoor use. Both types of units are available with a variety of circuits to service more than one compressor.

For further information, request Form No. 7011B, or call the Dunham-Bush sales engineer near you.



Horizontal Discharge Blower Condenser

Vertical Discharge 'LSBC' Blower Condenser



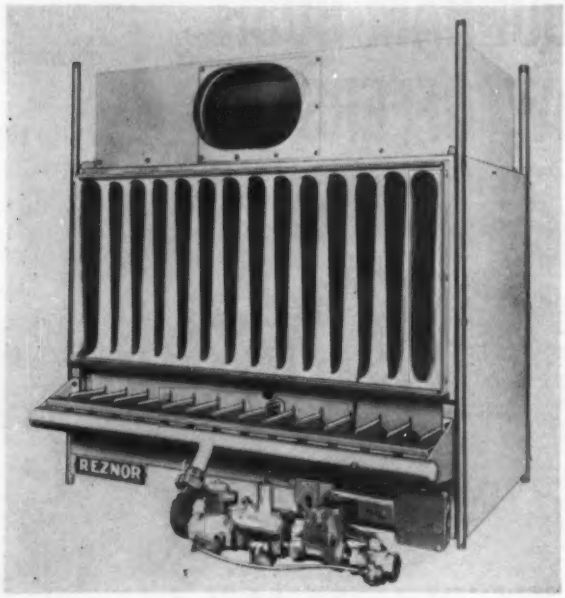
DUNHAM-BUSH

DUNHAM-BUSH, INC.

WEST HARTFORD 10, CONNECTICUT, U. S. A.

SALES OFFICES LOCATED IN PRINCIPAL CITIES

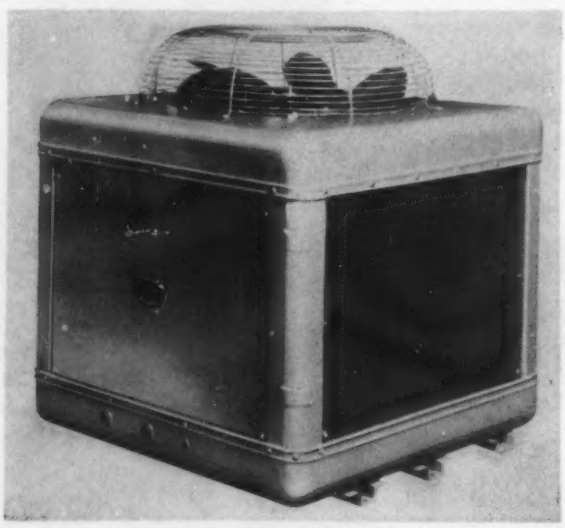
with drain connections. Units are available optionally as dual fuel models, allowing provision for use



of propane gas as standby fuel. Normally equipped with 115-volt controls, the furnaces may be converted to 24-volt operation and 230-volt wiring.
Reznor Manufacturing Company, Mercer, Pa.

AIR CONDITIONING EQUIPMENT

Two new units, a heat pump and a remote air-cooled fan-type condenser (shown) have been added to this line. Model 3THP-1, a self-contained, air-cooled heat pump, is a compact central heating and cooling unit in one package and is available in three ton. For mounting in an attic, crawl space, basement or on a roof, it can be installed inside, half-inside or outside, and no plumbing connections are necessary except for the condensate drain lines. Provided with the unit are a six-kw strip heater, heavy duty compressor which

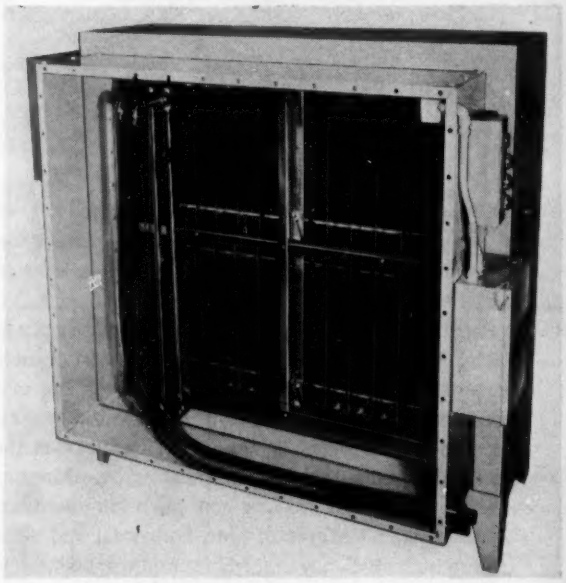


will operate at 130 F ambient temperature, propeller-type condenser fan, double drain pan and refrigerant circuit protection.

Designed for use with evaporators in air conditioning installations using upflow, downflow or horizontal flow furnaces, the condensing units are suited for new or add-on commercial or residential applications because of their compactness and economical installation. Units may be located outside the building or on a roof. Cooling coil fits on top of the furnace and is connected to the refrigeration unit with small copper tubing. Two and three-hp models are offered.
Gaffers & Sattler, 8111 W. Beverly Blvd., Los Angeles 48, Calif.

ELECTRONIC AIR CLEANER

Complete with front and back duct connections, drain pan and pre-assembled washing and oiling mechanism, the Type HVSC Electro-Cell Air Filter is designed for use in inaccessible areas and for use with air handling units and is available for either floor or ceiling mounting. Twenty-four sizes range from 1000 to 25,000 cfm. Automatic washing schedules can be set to meet the needs of individual jobs; design of the unit eliminates



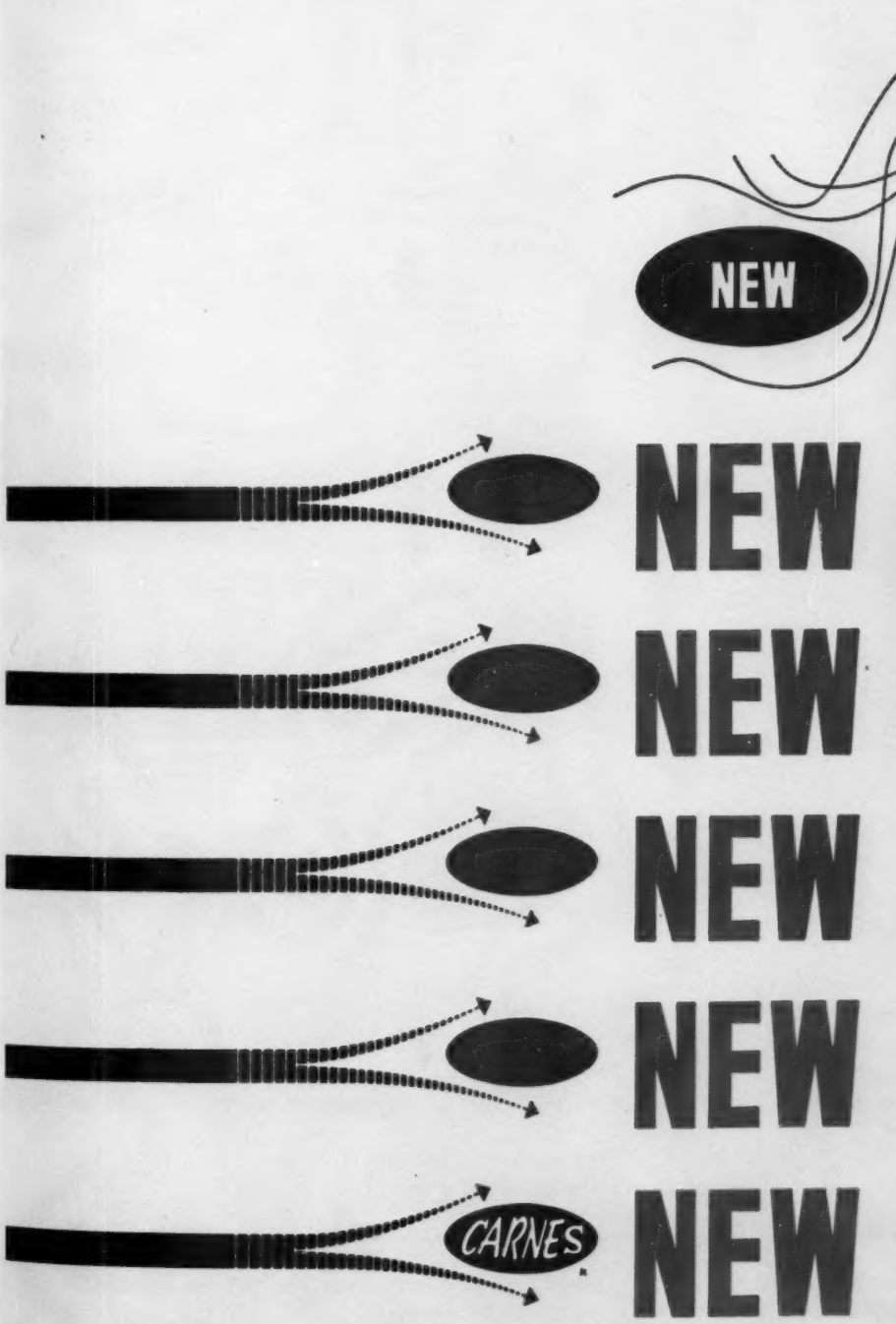
by-pass areas within the collector cell; precipitator face area and velocity are designed for coil face area and velocity; all insulators are outside the air stream; and ionizing wires extend the full height of the collector cell.
American Air Filter Company, Inc., 215 Central Ave., Louisville 8, Ky.

HEATER LINE

Expansion of the pool heater line into new Series II models ranging from 800,000 to 2,400,000 Btu/hr extends application of this line to large commercial swimming pools. Incorporated in the units are removable bronze headers for easy cleaning and descaling at either end of one-in. extruded copper fin tubes and heavy-duty, deep drawn copper tube sheet liners, a four-pass heat exchanger for maximum heat

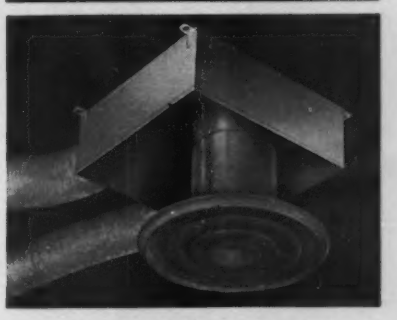
Carnes is first to offer engineers and contractors a completely packaged low-velocity control unit specifically designed to meet the requirements of dual duct low-velocity systems. Carnes Low-Velocity ATC Units . . . where limited space does not demand the smaller duct size of high-velocity systems . . . offer all the advantages of individual room control inherent in dual duct air distribution systems. 2 ATC Ceiling Models — end discharge and bottom discharge in 5 sizes. 2 ATC Window/Perimeter Models — front and top discharge in 2 sizes.

LEADERS GO TO CARNES FOR THE NEWEST IN AIR DISTRIBUTION



NEW

Dual Duct Low-Velocity ATC Units



NOW DUAL DUCT COMFORT AT LOW-VELOCITY COST

Carnes Low-Velocity Acoustic Terminal Control Units — ATC — provide "pinpoint" temperature control for specific zones through the flexibility of dual duct air distribution without the need for sealed ducts, high pressure fans and high fan horsepower. Carnes Low-Velocity ATC Units proportion the hot and cold air in response to individual room thermostats . . . blend the hot and cold air for discharge to individual space areas . . . attenuate self-generated noise plus duct-carried noise . . . provide manual volume adjustment.

For complete information, write for Carnes Low-Velocity ATC Catalog 1361.

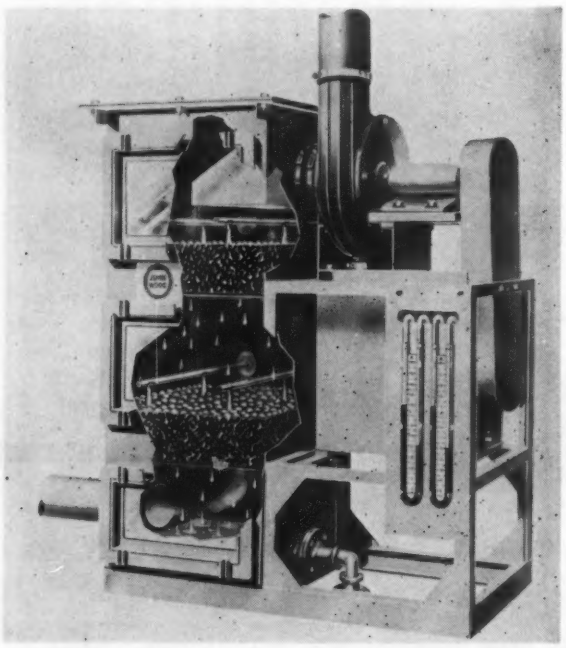
Carnes also manufactures: High-Velocity ATC Units • Modular, Round, Rectangular, Square, Convertible and Baseboard Diffusers • Registers • Grilles • Roof and Wall Ventilators.

CARNES CORPORATION, VERONA, WISCONSIN, / CANADIAN SUBSIDIARY: WEHR AIR EQUIPMENT COMPANY, LTD., 86 GERTIE STREET, WINNIPEG 2, CANADA

transfer and scale protection through velocity flow. An automatic temperature control regulator keeps the heater discharge temperature between 105 and 115 F.
Raypak Company, Inc., 2416 Chico Ave., El Monte, California.

FLOODED-BED SCRUBBER

Engineering refinements are cited as having improved the capability of this unit to remove extremely toxic and corrosive contaminants. Rated capacities avail-



able range from 200 to 5000 cfm. Utilized to eliminate fumes, mists, vapors and light dusts from processing gases are two packed beds, each flooded with a scrubbing liquid, which sustain continuous contact between the contaminated gas and the scrubbing medium. Concentration of molecules at the interface between the gas and liquid is intensified by the high pressure relationship between the gas and liquid phases.
John Wood Company, Air Pollution Control Div, Nicolet Ave., Floral Park, N. J.

DRIP TRAP

Principal use of this trap is for draining condensate and venting air in heating and low pressure systems such as unit heaters and ventilators, return ends of blast coils, hot water heaters and steam main and riser drips. Models are offered with a choice of ¾, 1, 1¼, 1½ and 2-in. connections, with maximum pressure of 15 psig. Traps can be supplied with the thermostatic air by-pass or without the thermostatic element and air by-pass for use where manual venting or external air by-pass is specified. Capacities of the units range from 50 to 5750 lb of water per hr.

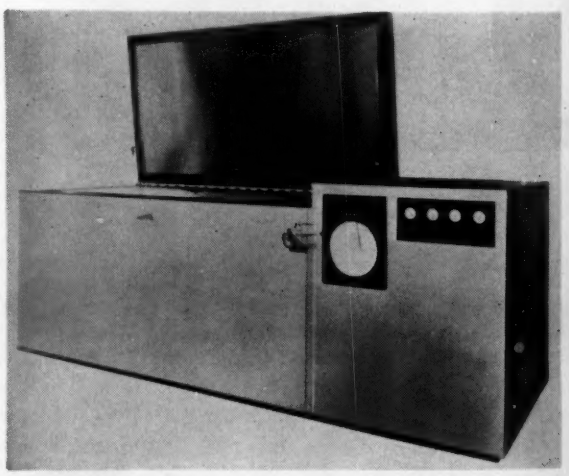
Features cited are: stainless steel water valve and seat, increased full-flooded water capacity, small diameter Sylphon bellows with stainless steel valve and seat for automatic air vents and all interior parts attached to cover and removable from trap body without disturbing pipe connections.
V. D. Anderson Company Div, International Basic Economy Corporation, 1935 W. 96th St., Cleveland 2, Ohio.

INDICATOR VALVE

An outside indicator on this valve allows the orifice position to be seen easily, even from a distance. Located outside the valve packing, the threads never come in contact with the flowing medium, preventing corrosion and galling. With O-ring type packing, interior dead space is minimized. Available in sizes from ⅛ to 4 in., the valve has a variety of standard tips and seats for various flow requirements. Pressure ratings are to 10,000 psi working pressure. The unit is offered in either manual or air actuated styles.
Flow Systems, Inc., 413 Poinsettia St., P. O. Box 444, Corona Del Mar, Calif.

LOW TEMPERATURE CHAMBER

Capable of reaching –250 F, Model Wt-8-250 utilizes a mechanically refrigerated closed system and was designed to make possible new applications for low temperature work in research, processing and production. Applications cited for the chamber include increasing the life of processing dies, tools, drills and



broaches; space vehicle component testing; blood storage; gas dehydration; and liquefaction of gases, among others.
Webber Manufacturing Company, Inc., P. O. Box 217, Indianapolis 6, Ind.

DENVER MEETING REAFFIRMED BY MEMBERS THE LABORATORY STAND OF DIRECTORS PAGE 43

ASHRAE SOLAR HEAT GAIN PROJECT REACHES PLASTICS-DOME SKYLIGHT PAGE 51

News highlights of the month

TRENDS

Milestone For the first time in its history, the American Society for Engineering Education has achieved a membership of 10,000. Founded in 1893, ASEE is the only professional society in its field established solely to study and improve education.

York expanding According to York President Joseph B. Elliott, more than $500,000 will be spent during the next few months in expanding the engineering research and testing facilities at the York Division of Borg-Warner Corp.

Safety survey Results of the tenth annual Safety Survey of the National Association of Refrigerated Warehouses reveal a reduction in the frequency rate in 1960, representing the second lowest on record. While the severity rate increased over the previous year, it remains the third lowest on record, it is indicated. A total of 173 plants reported; of this number 82 cited 3,194,039 manhours worked without an accident, a new record.

BOOK REVIEWS

Higher investment for science If the trend for increasing numbers of young people to want to become scientists is maintained, there will be twice as many scientists in 1970 as there are now, reveals a report, "Investing in Scientific Progress," which analyzes science education trends over the past 40 years and projects these trends to 1970. During the fiscal year 1961, it is cited, the United States invested from all sources a total of $3 billion in colleges and universities for science and engineering education and for basic research. This investment must grow to at least $8.2 billion in the fiscal year 1970, it is stated, so that every young person who shows the desire and capacity to become a scientist or engineer will have the opportunity to do so. Published by the National Science Foundation, 1951 Constitution Ave, N.W., Washington 25, D. C.

API report Marketing Division of the American Petroleum Institute has released a report on research and development needs for oil burning equipment, including information on the competitive situation in oil heating, the technical and economic aspects of residential oil heating equipment, the need for new design concepts and the action required for market retention and acquisition. Prepared by Battelle Memorial Institute in 1960, API Publication 1700, "Recommendations for an Industry-Wide Oil Burner Research Program" may be obtained from the Publications Section, API, 1271 Avenue of the Americas, New York 20, N. Y.; price, $5.

BRI proceedings Now available, "Performance of Buildings" contains proceedings of a conference held as part of the 1960 Spring Conferences of the Building Research Institute, Division of Engineering and Industrial Research. Copies of Publication 879 may be obtained from BRI, National Academy of Sciences-National Research Council, 2101 Constitution Avenue, Washington 25, D. C.; price, $5.

Automatic control Fundamentals, selection and application of automatic control as applied in the heating, ventilating, refrigerating and air conditioning field are covered within "Automatic Control of Heating and Air Conditioning," Second Edition, by ASHRAE Presidential Member and Fellow John E. Haines. Well illustrated, the book covers major types of heating, ventilating, refrigerating and air conditioning separately. McGraw-Hill Book Company, 330 West 42nd Street, New York 36, N. Y.; price, $10.50. (To be published in September.)

For electricians "1540 Questions and Answers for Electricians" contains questions from actual examinations taken by professional electricians on various levels of competence. Questions are grouped in chapters; sixty-three groups of questions are included. Arco Publishing Company, 480 Lexington Avenue, New York 17, N. Y.; price, $3.

Fluid mechanics

Maintaining the original organization of previous editions, "Elementary Fluid Mechanics," Fourth Edition, by John K. Vennard reflects the stronger emphasis currently on basic science in engineering education and a trend toward mathematical rigor, broader concepts and more acute understanding. Among the more significant changes in this latest edition are: more stress on flow field concepts requiring use of the partial derivative, a deeper treatment of compressible flow and a new chapter on some elementary concepts of mathematical hydrodynamics. John Wiley & Sons, Inc., 440 Park Avenue South, New York 16, N. Y.; price, $7.95.

Status of Russian engineers

Recent achievements and future goals of the Soviet Union in the field of engineering and engineering education manpower utilization are presented in "The Training, Placement and Utilization of Engineers and Technicians in the Soviet Union." This report is based on a tour of the Soviet Union made last year by six representatives of United States engineers sponsored by Engineers Joint Council and the National Science Foundation under an exchange agreement between the U.S. and U.S.S.R. The 112-page report is available in limited numbers for $1 from EJC, 29 West 39th Street, New York 18, N. Y.

Temperature measurement

More than 500 references to the field of temperature measurement are included within "Bibliography of Temperature Measurement" by Carl Halpern and Robert J. Moffat, National Bureau of Standards Monograph 27. References, covering the period from January 1953 to June 1960, with some earlier citations, are divided into a number of categories based on type of instrument used. Available from the Superintendent of Documents, U. S. Government Printing Office, Washington 25, D. C.; price, 15 cents.

Research and development

According to the National Science Foundation, colleges and universities spent $154 million during 1958 for research and development purposes in the natural and social sciences. Approximately three-fourths of this came from non-Federal sources; the remainder was contributed by the Federal Government. These data are reported in "Reviews of Data on Research & Development," No. 28, which may be obtained from the Superintendent of Documents, U. S. Government Printing Office, Washington 25, D. C., for 5 cents.

Handling frozen food

Published by the Cooperative Extension Service, University of Massachusetts in cooperation with the U. S. Department of Agriculture, "Handling and Merchandising Frozen Food" by F. Miles Sawyer, Thaddeus F. Midura and Richard M. Vondell was prepared for the use of State Extension Service personnel to assist in implementing training programs to improve the handling and merchandising of frozen foods. Intended to serve as a source of subject matter for such programs, the major emphasis is on retail handling. Free copies may be obtained from Food Distribution Program, Cooperative Extension Service, Stockbridge Hall, University of Massachusetts, Amherst, Mass.

Published by BRI

Those various publications available through the Building Research Institute are listed in the pamphlet, "Publications–April 1961". BRI, National Academy of Sciences-National Research Council, 2101 Constitution Avenue, Washington 25, D.C.

SPECIAL MEETINGS

Foreign affairs

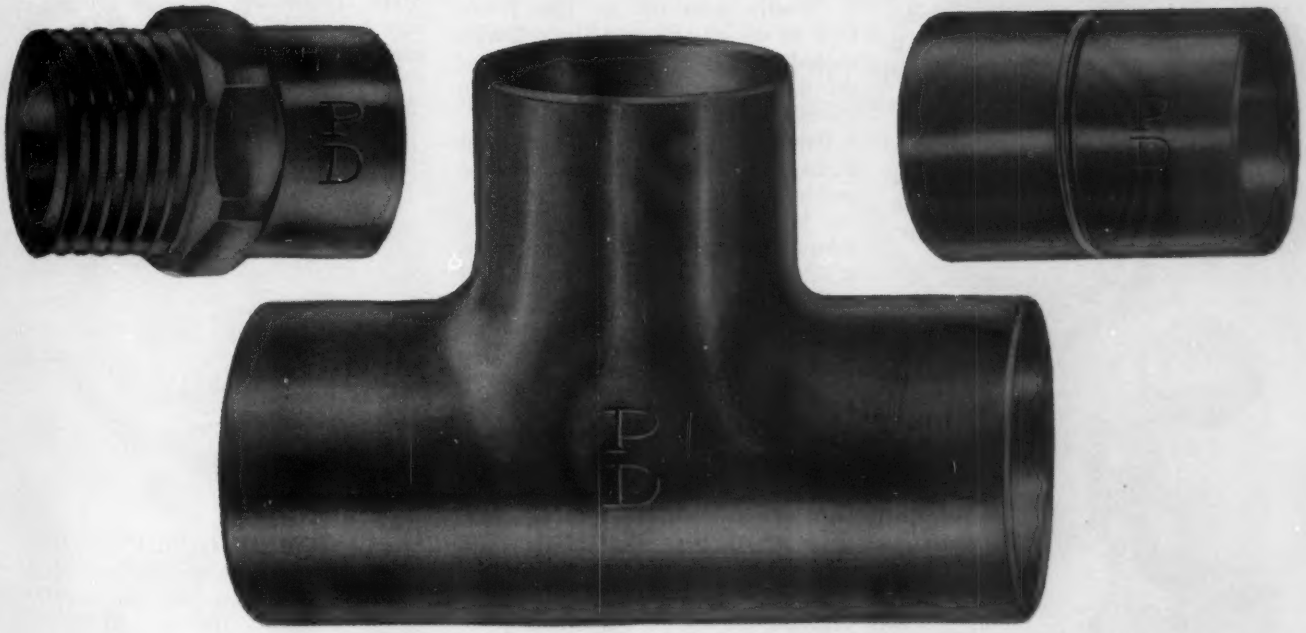
Engineering and world affairs was the theme of the 69th Annual Meeting of the American Society for Engineering Education, which included over 150 conferences, meetings and sessions, held at Lexington, Ky., June 26-30. Among other topics covered were: new fields of engineering to be incorporated into college curricula; research on the most effective ways of teaching engineering subjects; and administrative problems connected with teaching and research activities. Merl Baker, Member ASHRAE, was general chairman of local arrangements and participated in various sessions. Presidential Member R. C. Jordan was one of the speakers.

ECPD to meet

Engineers Council for Professional Development will hold its 29th Annual Meeting, October 2-3, in Louisville, Ky. The Engineering Education Session on October 3 will consider the topic – Where does professional specialization belong in the engineering curricula?

New addition to the famous Mine-to-Market Quality Line

PHELPS DODGE WROT FITTINGS for Air Conditioning and Refrigeration Tube



Phelps Dodge has expanded its line of products for the air conditioning and refrigeration industry to include a wide selection of wrot solder-type fittings. In keeping with modern specifications, both fittings and tube can now be obtained from Phelps Dodge—a single dependable source of supply.

PD wrot fittings are available in sizes ranging from ⅛-inch to six inches. All fittings are protectively packaged to assure factory-clean condition.

Initial warehouse stocks have been distributed to five key centers, Boston, Chicago, Los Angeles, Houston and Tampa, Fla., for fast, convenient service. Order through your Phelps Dodge wholesaler.

A copy of the new PD fittings catalogue is ready for you. Use the coupon below.

Phelps Dodge Copper Products Corporation, Dept. WF-1
300 Park Avenue, New York 22, N. Y.

Please send me your new fittings catalogue.

NAME_____

FIRM_____

STREET ADDRESS_____

CITY_____ STATE_____

Quality tube and fittings sold the quality way—through authorized refrigeration wholesalers!

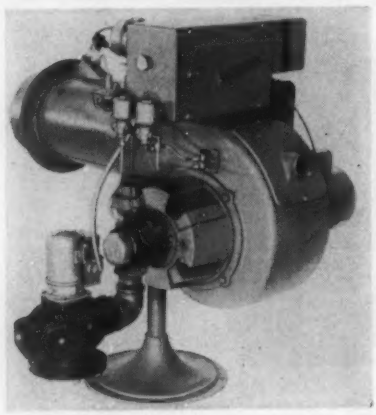
PHELPS DODGE COPPER PRODUCTS
CORPORATION
300 Park Avenue, New York 22, N. Y.

PARTS AND PRODUCTS

GAS-OIL BURNER

Input ranges from 700,000 to 2,250,000 Btu/hr and 1,400,000 to 4,200,000 Btu/hr, respectively, are provided by the two sizes introduced as Model J Cyclonetic burners. Each model is offered as a single or dual-fuel burner for any fuel gas or No. 2 fuel oil, arranged for either pedestal or suspended flange mounting.

Included in the standard burner package are an indicating program timer to sequence the blower motor,



flame rectification combustion safeguard and motorized gas valve or solenoid oil valves. The system provides for complete purge prior to pilot ignition and after normal operation.

Midland-Ross Corporation, Webster Engineering Div, 1402 S. 69th East Ave., Tulsa, Okla.

TEMPERATURE CONTROL

Conditionaire, an electronic temperature control system, operates without any shielded wiring, and there is no internal field wiring on the central control unit. Power actuators are driven directly by transistors, eliminating relays required in other systems. A major feature is the modular plug-in concept, which reduces maintenance costs and non-operating time. Malfunction of a circuit module is repaired by substitution of a spare circuit.

Fischbach and Moore, Inc., Controls Div, 646 N. Beachwood Dr., Los Angeles, Calif.

AIR DRYER

Model 25OAD refrigerated compressed air dryer dehumidifies 25 cfm of 100-psig air to a dew point of −12 F (atmospheric) for pneumatic process and temperature control systems and for applications requiring a supply of dry air. Input to the dryer is directly from the air compressor tank at up to 150 psig. Input air is cooled to slightly above freezing and the moisture is removed through an automatic discharge trap.

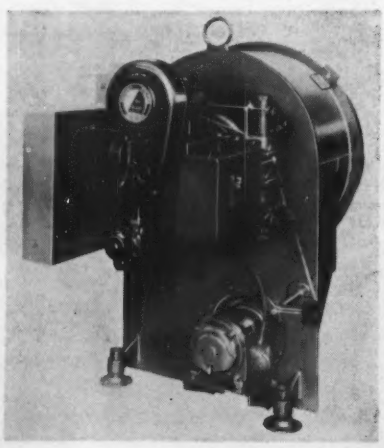
Featured is a Thermal-Mass heat exchanger, in which the refrigerant and air coils are surrounded by a special heat conductive material of considerable mass. Because of this mass, a substantial flow of compressed air through the exchanger results in but a small temperature rise in the exchanger medium. Operation of the unit is on 115-volt, 60-cycle current.

Zeks Industries, Inc., Edgemont, Pa.

COMBUSTION SYSTEM

Compact, fully integrated and packaged, the Forced Draft Thermopak will convert commercial or industrial high or low pressure boilers, ovens or dryers to automatic operation, utilizing fuel oil, gas or combination oil-gas. Intended for applications where natural draft is inadequate or induced draft unavailable, the unit may be applied to any firebox where pressure does not exceed ¾ in. wc at maximum firing rate.

Burner and accessories, fan, motor and control cabinet are mounted on

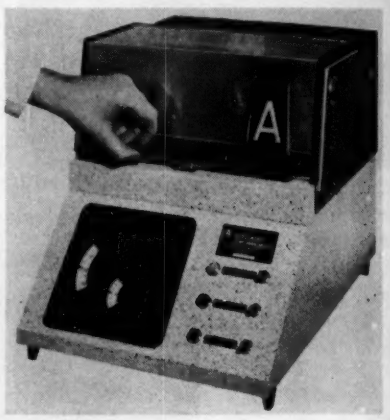


the extension front, permitting unobstructed access to boiler doors and cleanout areas. Oil supply requires but two simple connections and gas piping but one. Secondary air for combustion is supplied by an inset multiblade fan. The unit is available with four different models of horizontal rotary oil burners, a nozzle mix type gas burner or a combination oil-gas burner in ten sizes with maximum capacity from 15 to 175 gph for oil and 2200 to 25,500 cfh for 1050-Btu/hr gas.

Preferred Utilities Manufacturing Corporation, 41 E. 42nd St., New York 17, N. Y.

SMALL TEST CHAMBER

Mark II, a small, high-low temperature test chamber, has been added to this line of environmental test equip-



ment. Utilized is liquid CO_2 refrigeration. Internal working dimensions are 11 x 12 x 5 in. and temperature range is from −100 to 350 F. Included are a resistance element heater, aluminum liner and a fan with external blower motor for circulation.

Associated Testing Laboratories, Inc., Wayne, N. J.

FLOW-RATE CONTROLLERS

For purge and other flow control and indication applications, the Series 8800 line of controllers is completely self-contained. A single unit includes the rotameter, mounted on a needle valve-operated flow controller. An internal diaphragm assembly is designed to give a constant pressure differential for any valve setting. Several sizes are available, ranging in flow capacities from 0.01 gph (water) or 0.2 scfh (air) to 12 gpm or 48 scfm. Pressure rating depends on materials of construction: 250 psig for brass units, 500 psig for 316 stainless steel models. Diaphragms can be furnished in any of several material combinations.

Brooks Instrument Company, Inc., Hatfield, Pa.

DUCT INSULATION

Cited as being 2½ times stronger and 3 times more rigid than the company's previous duct insulation, E-Z Ply is a resin bonded, mineral fiber material with a breaking strength

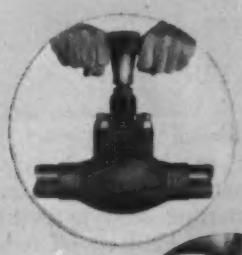
HENRY VALVE CO.

CONTROLLING FLOW SINCE 1914

WING CAP SHUT-OFF VALVES

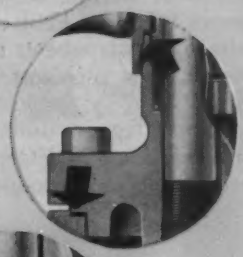
Globe and Angle Types with Bolted Bonnet and Full Capacity Flow

Henry offers the most complete range of sizes and types in the industry. Quality engineering, features of design, proven performance and adaptability to the most exacting specifications and diversified applications have made Henry Wing Cap Valves preferred by architects, consulting engineers, contractors and service men.



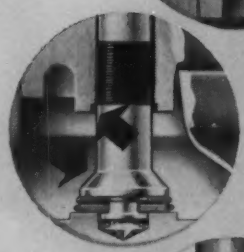
WING CAP SEAL

With Built-in Socket Wrench. For Ease and Convenience in Operating Valve Stem.



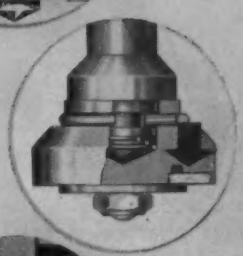
PROTECTIVE GASKETS

Fully Retained. Prevent Loss of Expensive Refrigerant.



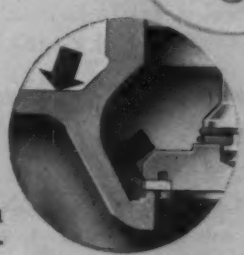
BACK SEATING STEM

Can Be Repacked Under Pressure.



NYLON SEAT DISC

Replaceable, Non-rotating, Self-aligning, Chatterproof.



HEAVY WALL & WEB

Choice of Bronze Alloy or Semi-steel.

Bronze alloy valves have integral O.D.S. connections. Ductile iron valves available with integral F.P.T. connections; flanged type available with brass O.D.S. or steel butt-weld adapters.

HENRY VALVE COMPANY

For Refrigeration, Air Conditioning and Industrial Applications

MELROSE PARK, ILLINOIS, U.S.A. CABLE: HEVALCO, MELROSE PARK, ILL.

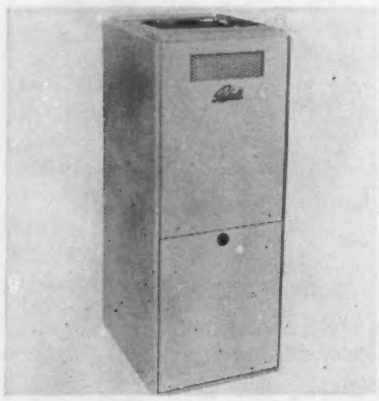
exceeding eight lb for a one-in. thickness six in. wide. The insulation is available either plain or with the following factory-applied facings: black Kraft paper, vapor barrier bonded with a coat of high penetration asphalt; scrim-reinforced aluminum foil, laminated with flame-resistant Kraft paper; embossed aluminum foil; and neoprene.

Thermal conductivity of the material is 0.24 at a mean temperature of 75 F and the maximum temperature limit is 450 F. E-Z-Ply is available in two by four-ft panels in thicknesses of 1, 1½ and 2 in. Light in weight, it can be applied by any of the conventional methods of installing semi-rigid duct insulation.

Baldwin-Ehret-Hill, Inc., 500 Breunig Ave., Trenton 2, N. J.

DUCTED FURNACE

One model is cited as providing installation flexibility for high-boy, counterflow or horizontal applications. Instant shut-off and return to minimum starting load in event of power failure or limit cut-off reduces wire size requirements and minimizes power surge problems. Controls are not affected by ambient temperature and each stage is positively sequenced and timed in fixed steps. Blower is belt-driven and separately controlled



to eliminate cold air at the register. Stage selection is available from 10 kw (34,200 Btu/hr) to 25 kw (85,500 Btu/hr).

Rybolt Heater Company, Ashland, Ohio.

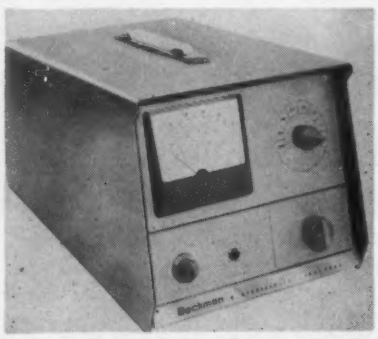
2½-TON CONDENSING UNIT

Refrigerant subcooling through special coil circuiting is cited as providing a high Btu/hr/watt performance in the new HSA2-251, nominal 2½-ton condensing unit. Liquid refrigerant reaches the evaporator up to 20 F cooler than the condensing temperature. Electrical and refrigerant connections are made at a central location on the condenser side of the cabinet, with flare fittings provided for the refrigerant plumbing. An axial-flow fan delivers 1855 cfm in a straight line pattern through the condenser, which has a 3.75-sq ft face area and four rows of ½-in. tubes. Standard accessories include a combination heating-cooling thermostat and auxiliary control box containing strips and relays for controlling evaporator blower, compressor and furnace.

Lennox Industries, Inc., Marshalltown, Iowa.

HYDROCARBON ANALYZER

Operating on the principle of hydrogen flame ionization of carbon atoms,

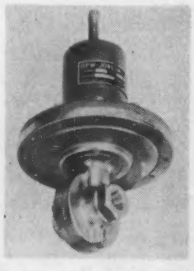


Models 108 and 109 detect and measure trace concentrations of hydrocarbons in gases, vapors or the atmosphere, providing a sensitivity of zero to four ppm carbon full scale. Model 108 is for panel mounting and Model 109 (shown) for laboratory or field use. Response is directly proportional to the sample injection rate and to the number of carbon atoms in the sample. Measurement is both in ppm and in percentages of concentration. Among applications cited are: controlling solvent and paint vapors in production plants, air pollution studies, determining muffler and afterburner effectiveness and checking combustion efficiency.

Beckman Instruments, Inc., Scientific and Process Instruments Div, Fullerton, Calif.

PRESSURE REGULATOR

Available from stock in ductile iron (Model 160A) or bronze (260A) in ¼, ⅜, ½ and ¾-in. sizes is a new, extra sensitive pressure regulator. Suitable for steam, water, air, oil, gas or chemicals, the unit has a pressure control range from 0.5 to 125 psi and is self-operated. Stainless steel, self-cleaning and self-lapping sliding gate seats are standard. Suitable for 250 psi at 500 F, the unit has a maximum pressure drop of 175 psi and a minimum drop of 3 psi.



OPW-Jordan Corporation, 6013 Wiehe Rd., Cincinnati 37, Ohio.

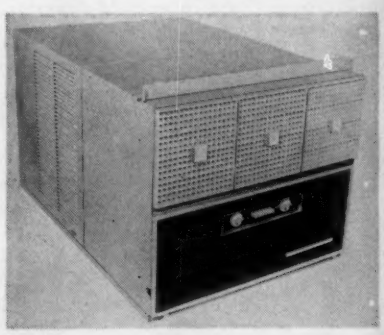
UPRIGHT FREEZER

New for supermarkets is a Chill-Air upright, reach-in freezer with a storage capacity of 172 cu ft. It will handle 6000 lb of frozen food at −10 to −15 F. Over-all dimensions of the unit, Model CAF 172, are 10 x 4 x 7 ft. Two and three-door versions are available. Exterior is finished in baked-on enamel over heavy-gauge steel, interior walls are of rust-proof aluminum and the entire unit is insulated with high-density glass fiber with an aluminum vapor barrier. A low voltage heating element in the doors keep them from icing.

Erickson Industries, Inc., River Falls, Wisconsin.

WINDOW CONDITIONERS

Four new Everest window air conditioners have cooling capacities of 24,000 and 28,000 Btu/hr and are capable of removing 4.5 and 7.3 pints of moisture from the air per hr, re-



spectively. All four models, operating on 208 and 230 volt, feature a built-in thermostat and exhaust and ventilation controls. Three-speed blower system and wide, unobstructed air passages permit fast cooling at a low noise level.

Fedders Corporation, 59-01 Grand Ave., Maspeth 78, N. Y.

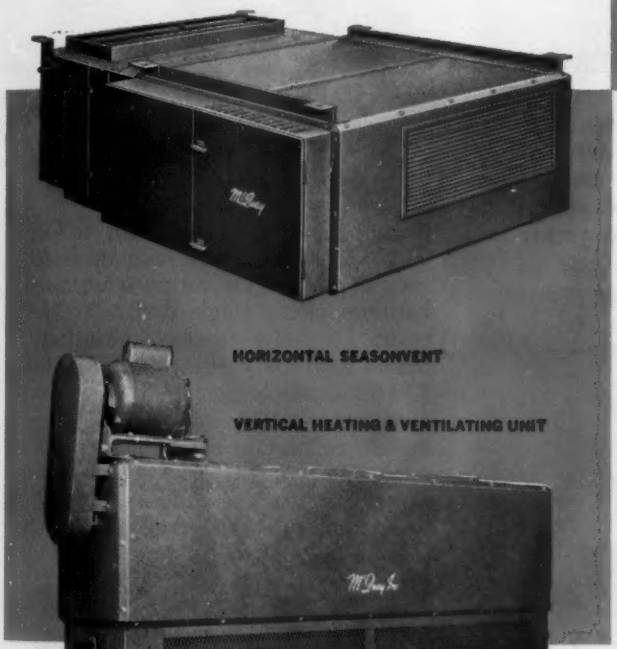
SOLAR HEAT CONTROL

Manufactured by installation of a preframed louver shade screen between two panes of glass that are sealed hermetically, Solarshade provides control of solar heat and sky glare in window and wall construction without maintenance. The screen is located permanently in the dead air space of

QUIET!

and flexibility at its very best!

VERTICAL SEASONVENT



HORIZONTAL SEASONVENT

VERTICAL HEATING & VENTILATING UNIT

McQuay HEATING AND VENTILATING UNITS

McQuay "HC" heating and ventilating units are the most versatile and flexible to be found anywhere with the widest range of coil selection in the industry, and with capacities for vertical and horizontal models up to 48,330 cfm in nineteen sizes. They are designed for horizontal, vertical, wall and inverted installations for use with or without duct work. They can be used with hot water or steam, with or without duct work, and will fill the needs of every installation.

The McQuay Seasonvent is engineered specifically for auditoriums and theatres where quiet operation is demanded. This large capacity heating and ventilating unit is also extremely versatile and flexible with nine sizes for low static applications (0" to 1¼" s.p.) and seven sizes for high static installations. Capacities range from 1,250 to 15,000 cfm. Contact your nearest McQuay representative for complete information, or write to McQuay, Inc., 1606 Broadway N. E., Minneapolis 13, Minnesota.

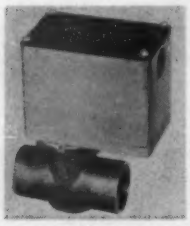
McQuay INC.

AIR CONDITIONING • HEATING • REFRIGERATION

the unit; the entrapped air is at atmospheric pressure.
Armour Glass Company, 8331 Isis Ave., Los Angeles 45, Calif.

ZONE VALVE

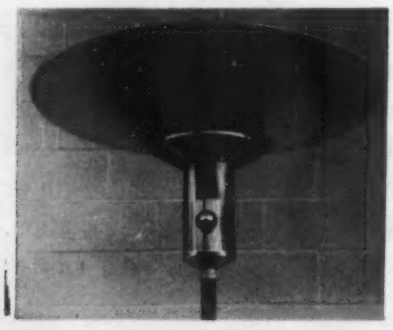
Thermostatically controlled and motor operated, this hydronic zone valve, designated VM-120, is for control of water or steam in residential and light commercial heating or cooling systems. A low voltage, wing-type for installation at the radiator or in the supply piping to the zone to be controlled, the valve is compact in design, permitting easy installation between rafters and joists or in convectors. Other features are synthetic packing which provides protection against leakage and a 24-volt electric motor requiring no lubrication.



Body rating of the unit is 50 psi max, and highest operating pressure differential is 15 psi. Electrical specifications are: two or three-wire, two-position; non-current failure.
General Controls Company, 801 Allen Ave., Glendale 1, Calif.

OUTDOOR HEATER

For outdoor applications, this gas-fired, infra-red radiant heater may be installed permanently, as a gas light, or on a portable base for either natural or propane gas. At temperatures to 1600 F, the stainless steel, ceramic-coated heat exchanger provides heat



for an area up to twenty ft diam. Input is 50,000 Btu/hr.
Solarflo Company, Inc., Box 745, Glendora, Calif.

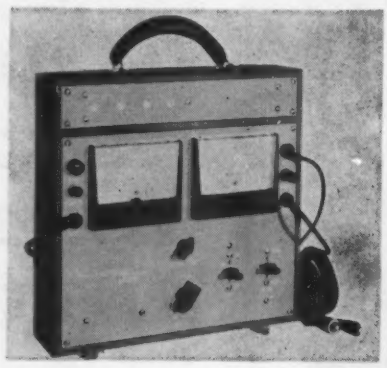
COMPACT CONDITIONER

Small dimensions characterize the RAC-1, Series 4, air-cooled remote compressors for residential and light commercial air conditioning. First of the series to be introduced is Model 30, rated at 30,000 Btu/hr at 95 F; it is to be followed by the five-ton, 60,000-Btu/hr Model 60. But 16⅞ in. wide, the unit is 25-13/32 in. high and 44½ in. long.

Slim width has been achieved by a side intake, top discharge pattern of handling condenser air. Condenser coil is a 5.53-sq ft, three-row design with ½-in. tubes and 13 rippled surface aluminum fins per in. Liquid refrigerant sub-cooler and liquid sight-glass with moisture indicator are included. The fan is a 16-in. propeller-type, powered by a 1/6-hp, 1050-rpm motor.
Payne Company, 855 Anaheim-Puente Rd., La Puente, Calif.

INSULATION TESTER

Measurement of insulation resistance with dc test potential adjustable from zero to 2500 volt is provided by the Model 2955 Vibrotest. Designed for dielectric absorption tests and plotting of insulation resistance versus time



curves, the instrument has a dc test potential continuously adjustable over the complete range, allowing comparison of insulation resistance at any selected voltage. An electronic power supply, operating from self-contained batteries, provides the high voltage test potential. Separate 4½-in. scale meters indicate test voltage and insulation resistance over the range of 100,000 ohm to 50,000 megohm.
Associated Research, Inc., 3777 W. Belmont Ave., Chicago 18, Ill.

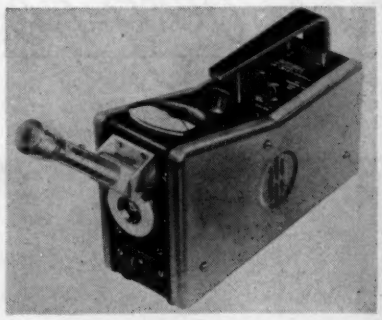
COUPLINGS

Featured in a new series of cryogenic and high temperature couplings is an all-metal seal, cited as maintaining a tight joint for extremes of temperature shock, pressure surge and vibration. Several configurations are available in aluminum, stainless steel and other materials, to accommodate all types of fluids at any desired level of pressure from −425 to 1800 F. Sizes range from ¼ through 3 in. tube diam.
Gamah Corporation, 1654 Lincoln Blvd., Santa Monica, Calif.

THERMODYNAMIC INVESTIGATION ON THE PERFORMANCE PLANE FOR A REFRIGERANT EXPANSION ENGINE PAGE 75

SOUND LEVEL METER

Frequency range of the Model 412 is five cps to thirty kc. A ceramic micro-



phone allows the instrument to be used in temperatures ranging from zero to 170 F. Operation is on two batteries plus a single bias cell. Circuitry includes seven transistors. Sound level range of the unit is 24 to 150 db. Controls include main attenuator, meter-battery switch, weighting switch, calibration potentiometer and microphone equalizing switch.
H. H. Scott, Inc., Instrument Div, 111 Powdermill Rd., Maynard, Mass.

PACKAGE GENERATOR

Added to this line is a two-pass Scotch package generator with a rated output through 600 hp. Available in both a wet back version producing either steam or hot water and in a dry back model for steam process work, the 202 Series can be fired with gas, oil or gas-oil in combination. Heating for large commercial and industrial buildings is provided by the low pressure model, the 202L; the high pressure unit, 202H, is designed for 100, 125 and 150-psig steam, and is used normally in commercial-industrial process work.

Each contains five sq ft of fireside heating surface per hp and reduces draft loss to a minimum, resulting in a low noise level in the combustion blower. Since each utilizes forced draft, there is no need for an induced draft fan or a high stack. The low pressure model has a full wet back rear furnace and the high pressure model is of dry back design, featuring a specially designed, refractory-insulated rear door that can be swung open on a hinge for full access.
Iron Fireman Manufacturing Company, 3170 W. 106th St., Cleveland 11, Ohio.

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Comment

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LET'S GO TO THE MOON

Not so long since, we came across a lengthy and learned study and defense of why the World must, with the utmost expedience, succeed in sending men to the Moon, to Venus and even beyond.

It was one of these casually plausible things that started with the endeavors of Pericles to provide such projects as the youths of Athens might rally round so many centuries ago for the edification of posterity and ended with magnificent disdain for such questions as money, time or effort as being literally so unimportant as to lie beneath serious consideration.

We think it is wonderful to be a Cloud Niner, or a Torch Bearer, or any other kind of starry-eyed causist if that is the way one wishes to fill the gap in time between Youth and Age, but how some folks can learn so much from History about some things and so little from the same source about others is completely beyond us.

Inflation of the currency, overspending, and disregard for debt have brought ruin to innumerable reigns and nations; we have yet to hear of one which spent its way to prosperity and survived economically to enjoy it.

THE PRICE TO BE PAID

The French, who know everything—except what to do about it—have two wise sayings among their many aphorisms. One, "The only lesson of experience is that, given the same opportunities we shall make the same mistakes" and the other "The lesson of experience is that there are no lessons of experience". Take your choice; they both apply richly to our present crises.

For people never learn; note, we did not say individuals.

Generation after generation bumbles against the same imaginary walls and barriers, lacking the courage or initiative to walk straight through them.

Nation after nation comes up to the same economic impasse and, facing the inevitable "uncomfortable" solution, rides the road to ruin by inflating its economy.

Civilization upon civilization has fallen the victim of its own shortcomings and imaginings long before it succumbed to the hungry jackals outside its boundaries.

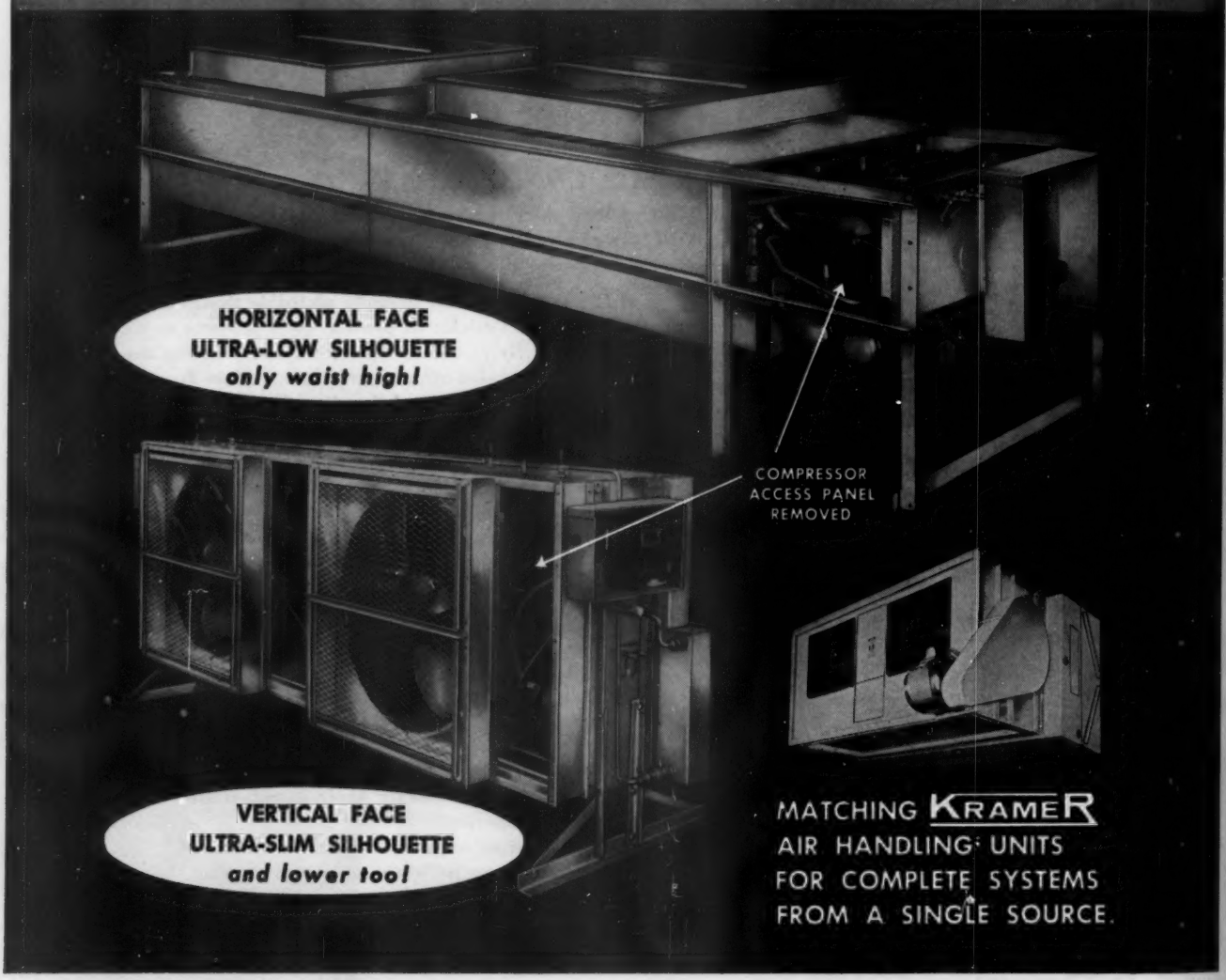
Must we go that way, too?

Edward R Searles
Editor

From the leader in air-cooled condensing

the new KRAMER UNICON-COMPRESSOR

PACKAGED *WATER-FREE* CONDENSING UNIT



3 to 70 TONS—HORIZONTAL or VERTICAL FACE

The low, slim-trim lines of the new Kramer air-cooled UNICON-COMPRESSOR give it unequaled flexibility in any architectural setting. Space saving outdoor design and low, low operating weight combine to reduce structural and engineering problems. Easy and economical to install, the complete packaged UNICON-COMPRESSOR has matchless accessibility for servicing. Its corrosion resistant aluminum casing, and its frame galvanized after fabrication, eliminate painting maintenance.

And remember—no place in the U. S. A. is too hot (or too cold) for the UNICON-COMPRESSOR. With the patented Kramer Winterstat it will operate any time of the year, without adjustment, even in the dead of winter.

Write for bulletin C460B.

KRAMER TRENTON CO., Trenton 5, N. J.

47 YEARS OF CONTINUOUS ACHIEVEMENT IN HEAT TRANSFER

How face air velocity affects

Airplane Passenger Comfort

E. J. OVERMYER
Member ASHRAE

As travel by air increases so does the need for passenger comfort.

Many airplanes now are equipped with individual air outlets which can be controlled by the passenger at his seat. This paper explores the effect of face air velocity on passenger comfort—based on findings during studies made by the author's company to determine a means of increasing outlet air velocity.

Many airplanes now in service have air distribution systems that allow each passenger to control a relatively high velocity stream of air delivered to his seat area. The passenger can control a small outlet which may be opened, closed or adjusted to suit individual needs. This outlet also can be swiveled to direct air on the face or other parts of the body. These outlets are referred to as passenger air vents, individual air outlets, "gaspers" or "eyeballs." They are helpful in that they enable the passenger to obtain a temperature lower than that of the general cabin, aid in smoke clearance while he or people adjacent to him are smoking, and furnish an added fresh air source in relief of air sickness.

Design of the individual air outlet system in Boeing 707 jet transport planes was based on the assumption that one-third of the outlets would be open, with a velocity of 1000 fpm 18 in. away as the objective. This objective essentially was met. We found in practice, however, that there were many times when all of the outlets were open, and at these times the velocity was lower than desired. Studies were conducted to determine means of reducing substantially the pressure drop in the distribution ducting, but any major change involved changing other things in the airplane also, and it was concluded that this was not practical.

An alternate method of increasing the discharge velocity was to install a fan in the duct system to boost the air pressure. In our 707s, with vapor cycle refrigeration, there are recirculation fans which provide air flow on the ground. Turbo-compressor or engine bleed air is used in flight. However, in those of our 707s which have air cycle refrigeration there may be extended periods of time on the ground when the air flow is zero or quite low. The procurement and installation of a gasper boost fan would have the added advantage of providing individual outlet air flow under this condition, which the improvements in duct pressure loss alone would not do.

We knew, of course, that the fan would add heat to the air, and that the more we increased the velocity the greater the fan power requirement, the greater the fan size, and the more heat would be added to the air. We did not know what value of air velocity at the passenger's face would be high enough to satisfy nearly everyone, although we felt that localized air velocities have a different effect on comfort than velocity over the entire body. Also, we did not know to what extent the air temperature rise would offset the comfort gain of increased velocity. To answer these questions, we decided to undertake a test program in which a large number of people would be subjected to varying conditions of individual air outlet velocity and temperature, varying cabin ambient temperatures and varying humidity.

Test Procedure—Test subjects were seated in airline type seats within a mock-up of a 15-ft section of the passenger cabin. The outlets were adjusted so that the air was directed against the subject's face and were not moved throughout the test. At each cabin ambient temperature the subjects were given about 10 min for acclimatization. They then were subjected to varying conditions of gasper velocity and temperature as shown in Table I. After each condition was set up the subjects were asked to

E. J. Overmyer is a Research Engineer with Boeing Airplane Co.

indicate their reactions as to general comfort, comfort with respect to the previous condition, temperature of the air from the gaspers and gasper air velocity.

DISCUSSION OF RESULTS

Table I lists the test conditions and shows the percentage of people feeling most comfortable at each of a set of conditions for numbers 1a through 9b, and the number of people feeling most comfortable for conditions number I-1 through II-10. Tables II, III and IV show the number of people checking each reaction at the various conditions of cabin ambient temperature and humidity, individual air outlet velocity and discharge air temperature.

An examination of the tables shows that, considering that if the velocity is too high the passenger can close down the outlet, 600 fpm will satisfy 100% of the people at a cabin ambient of 85 F and 20% R.H. and 95% of the people at a cabin ambient of 95 F and 15% R.H. A velocity of 700 fpm will satisfy 99% of the people at a cabin ambient of 95 F and 15% R.H.

The bulk of the testing was done at conditions designed to simulate those obtaining in a vapor cycle air conditioned airplane at an 85 F cabin and in an air cycle air conditioned airplane at a 95 F cabin. For the vapor cycle system a base of 365 fpm and 60 F was chosen as simulating what now exists with recirculation fans operating and full cooling from the packs on a hot day. A calculated temperature rise from the boost fan then was added for each corresponding increase in gasper air velocity. A similar procedure was followed for the air cycle system starting with a base of 70 F conditioned air and 0 gasper air velocity, which is roughly full cooling on the ground at engine idle. No humidity was added to the cabin ventilation air nor to the gasper air.

Additional testing was done with gasper air temperatures higher and lower than the above. No definite conclusions can be reached from these tests, but the trend seems to be towards comfort at lower air velocities if the gasper air temperature is low. This is reflected more in the comments received from the subjects than anything shown on the tables, although it can be seen from Tables III and IV that at 95 F cabin a large percentage of the subjects felt that the gasper air temperature was too hot at all conditions.

A group of eight subjects was exposed also to a set of conditions where the cabin air and gasper air humidity were increased to about 57%. This seemed to have an appreciable effect on the general comfort level (adversely), but did not affect the feelings with respect to gasper air velocity.

Vapor cycle air conditioning has the capability of maintaining an 85 F cabin on a hot day with a full passenger load and the doors closed. Under these conditions but 5% of the people felt that the present gasper air velocity was too low. The difficulty, of course, is that the doors are required to be open for passenger and galley loading. If the cabin temperature increases a higher velocity is required.

In all, 92 different people were subjected to the various conditions, some of them participating twice. Twenty-six of the subjects were women. The age of all subjects ranged from 18 to 55 with the

(*Continued on page* 44)

Table I

Condition Number	Cabin Ambient F	Relative Humidity %	Outlet Air Temperature F	Air Velocity at 18 in. fpm	Subjects Feeling Most Comfortable for Each Set of Conditions Percent of Total
1a	85	20	79	365	29
2a	85	20	82	500	17
3a	85	20	87	600	37
4a	85	20	93	700	17
5a	95	15	—	0	8
6a	95	15	88	365	12.5
7a	95	15	95	500	25
8a	95	15	99	600	42
9a	95	15	107	700	12.5
1b	85	20	60	365	30
2b	85	20	65	500	32
3b	85	20	69	600	17
4b	85	20	73	700	21
5b	95	15	—	0	11
6b	95	15	80	365	14
7b	95	15	85	500	21
8b	95	15	90	600	12
9b	95	15	96	700	42
					Number of Subjects
I-1	95	57	—	0	1
I-2	95	57	80	365	1
I-3	95	57	85	500	3
I-4	95	57	90	600	0
I-5	95	57	96	700	3
II-6	95	15	—	0	0
II-7	95	15	79	365	2
II-8	95	15	79	500	2
II-9	95	15	79	600	7
II-10	95	15	79	700	4

Table II

85 F Cabin and 20% R. H.

Air Velocity, fpm	365		500		600		700	
Outlet Air Temperature, F	79	60	82	65	87	69	93	73
Reaction	Number of People Responding							
Comfortable	15	45	13	48	18	46	18	46
Warm	9	14	11	11	6	5	5	13
Uncomfortable	0	0	0	0	0	0	0	0
Hot	0	0	0	0	0	0	0	0
Gasper Air Too Hot	0	2	0	1	1	0	0	2
Air Just Right	22	57	22	51	21	52	19	54
Air Too Cool	2	0	2	6	2	6	5	3
Air Velocity Too High	7	12	12	26	15	41	16	43
Velocity Just Right	14	45	10	31	9	17	7	16
Velocity Too Low	2	2	2	1	0	0	0	0

RICHARD C. JORDAN
Guest Sponsor

The IIR and ASHRAE

Half century ago ASHVE and ASRE, our two predecessor societies, between them totaled fewer than a thousand in membership. Far away in another part of the world the first International Congress of Refrigeration of 1908 was being held at the Sorbonne in Paris and this initiated the International Institute of Refrigeration. Two weeks of travel separated the two continents, and international radio transmission was only an experiment. The affairs of countries and continents beyond our shores seemed unimportant and of little real concern.

Today the world is faced with three explosions. The first, well recognized, involves population. In the last fifty years our own country has grown from one hundred million people to one hundred eighty million, and the world, from two billion to three billion, with a predicted population of five to eight billion within the next century. The second explosion is political. One-third of the world's population, eight hundred million people in twenty-two countries, have received their political independence in the last twenty years. Furthermore, the majority of these eight hundred million people live in countries where the annual income per person is considerably less than three hundred dollars per year.

This political revolt coincides with the dramatic breakthroughs in science and technology which constitute the third explosion. For the first time in the history of the world we have the knowledge and the power to eliminate hunger and elemental human want anywhere on earth.

Of late we have had dramatic proof that the sources of this explosion in science and technology are by no means limited to the North American continent. Any egocentric beliefs in our own universal technical superiority have been shattered by the

RICHARD C. JORDAN

achievements of the Russians, the Red Chinese and others. We have awakened to the fact that many of the fundamental technical contributions have originated beyond our own shores and that in the quickening pace of our present civilization we must participate in the world-wide community of scholars and engineers or lag behind. The international exchange of ideas and developments between, for example, New York, Paris, London, Rome, Calcutta and Tokyo is carried out to the advantage of all.

In recognition of this growing need for international participation, the Council of the American Society of Refrigerating Engineers in 1955 requested that the National Academy of Sciences-National Research Council form the U.S. National Committee for the International Institute of Refrigeration and apply for membership in IIR.

As a result of this and similar requests, the U.S. National Committee for IIR was born, composed of representatives of ASRE, ASHAE, ASME, the Refrigeration Research Foundation, the American Physical Society, the American Institute of Biological Sciences, the Institute of Food Technologists, the Air Conditioning and Refrigeration Institute, the American Institute of Chemical Engineers, the Cryogenic Engineering Conference and the American Chemical Society, together with representatives-at-large and representatives from the U.S. Departments of Agriculture, Commerce and the Interior. The National Academy of Sciences-National Research Council, on behalf of the U.S. National Committee for IIR, applied for membership, and in late 1956 in culmination of several years of negotiations the Executive Committee of IIR extended a formal invitation for membership. Following that date the U.S. National Committee for IIR has gained an increasing voice in IIR affairs and in the nine divisions or Commissions of IIR holds one presidency and four vice-presidencies, as well as vice-presidencies on the Technical Board and on the Executive Committee.

The work of the Institute is performed by nine specialized commissions ranging in interest from the scientific problems of low temperature physics and thermodynamics to refrigerated transport, education, air conditioning, design, construction and operation of refrigeration and air conditioning machinery, and applications of refrigeration to foodstuffs and agricultural produce. The membership represents every continent of the world with more than fifty country memberships from both sides of the Iron Curtain. The contributions reported through its members have been impressive and include, for example, the work of Joffé of Russia in thermoelectricity, de Haas and Gorter of Holland in the cryogenic field, Rudolph Plank of Germany in thermodynamics, and Fidler of England and Kuprianoff of Germany in food refrigeration, to name but a few. Of late some of the world's new nations, such as Tchad, Gabon and Mali, have joined IIR with a view toward gaining guidance in the refrigeration aspects of their own developing economies.

In August, 1962, the U.S. National Committee for IIR and its spon-

Dr. Richard C. Jordan, a Presidential Member, has been Chairman of the ASHRAE International Relations Committee for a number of years.

soring organizations will host meetings of five of the Commissions and of the Technical Board in Washington. Commission II, concerned with heat transfer and thermal properties and materials, Commission III, concerned with refrigeration and air conditioning machinery, Commission IV, concerned with food and agricultural applications of refrigeration, the air conditioning phases of Commission VI activities, Commission IX, concerned with education, and the Technical Board of the Institute of International Refrigeration will be involved in these meetings. Between one hundred and one hundred fifty top flight foreign engineers and scientists will meet with perhaps an equal number of American representatives and visitors to afford us with our first real opportunity to strengthen and enhance our international liaison in ASHRAE's interests.

ASRE, ASHAE, ARI and TRRF, the four groups which have been strongly instrumental in the fostering and initial development of the U. S. National Committee for IIR, can take pride in the growing importance of this rapidly developing activity. It is of mutual benefit to all nations to exchange information in the field of refrigeration and to keep aware of the scientific and technical advances which are occurring in other countries in order to accelerate advancement and prevent needless duplication of effort. It is of value to our own commercial interests to participate in the standardization of nomenclature and descriptive terms and in the drawing up of acceptable codes and standards as are now proceeding in other parts of the world. Above all it is both our privilege and our duty to participate in the establishing of working relationships with the world community of scientists and engineers with which we have common interests.

AIRLINE COMFORT

(*Continued from page* 42)

average age 35. Height of the subjects ranged from 4 ft 11 in. to 6 ft 2 in. with the average height 5 ft 8 in. The weight of the subjects ranged from 108 to 240 lb with the average weight 162 lb.

All of the subjects were company employees, which may have tended to make them somewhat less critical than if they had been outsiders. This drawback was recognized at the outset but it was decided to use them anyway for the sake of expediency. The trend of the reactions should not have been affected by this factor.

Twenty-two of the subjects had never traveled by commercial airliner, 25 had traveled 1 to 3 times, 23 had traveled 4 to 9 times, and 22 had made 10 or more trips by commercial airliner. Practically all of those traveling by commercial airline had had some experience with individual air outlets (gaspers).

About twenty of the subjects were engineers who may have had some preconceived idea of the re-results. However, an examination of their answers does not reveal any significant difference from those of the normal group.

Reactions of the subjects were also separated by sex, physical size, age and air travel experience; these group reactions were examined, also. Other than the fact that the taller people were satisfied with a lower air velocity as measured at the 18-in. distance, there was no significant variation.

Table III

95 F Cabin and 15% R. H.

Air Velocity, fpm	0	365		500			600			700		
Outlet Air Temperature, F	—	88	80	95	85	79	99	90	79	107	96	79
Reaction		Number of People Responding										
Comfortable	0	4	1	3	1	0	4	1	4	5	3	3
Warm	42	12	33	11	24	13	13	13	11	13	32	11
Uncomfortable	60	7	39	10	32	3	5	26	1	6	10	2
Hot	4	1	1	0	1	0	2	1	0	0	1	0
Gasper Air Too Hot		6	43	6	32	7	6	29	7	7	25	7
Air Just Right		18	31	18	24	9	17	29	9	17	32	9
Air Too Cool		0	0	0	1	0	1	0	0	0	0	0
Air Velocity Too High		2	0	7	7	2	11	21	10	19	33	11
Velocity Just Right		11	35	14	41	12	13	33	5	5	23	5
Velocity Too Low		10	39	3	10	2	0	4	1	0	1	0

Table IV

95 F Cabin and 57% R. H.

Air Velocity, fpm	0	365	500	600	700
Outlet Air Temperature, F	—	80	85	90	96
Reaction		Number of People Responding			
Comfortable	0	0	0	0	0
Warm	1	2	2	2	2
Uncomfortable	7	6	6	6	6
Hot	0	0	0	0	0
Gasper Air Too Hot		5	6	5	6
Air Just Right		3	2	3	2
Air Too Cool		0	0	0	0
Air Velocity Too High		1	3	3	4
Velocity Just Right		5	4	5	4
Velocity Too Low		2	1	0	0

CONCLUSIONS

The findings of this test program may be summarized by the following points:

- Adding a boost fan is a feasible method of improving the individual air distribution system performance.
- The objective for this system should be a minimum air velocity of 600 fpm measured at 18 in. from the outlet, with all outlets open.
- The passengers will feel comfortable at a lower face air velocity if: the cabin ambient temperature is reduced or the temperature of the gasper air is reduced.
- With elevated cabin relative humidity, the face air velocity does not have as much effect on increasing passenger comfort as when the cabin air is comparatively dry.

How periodic heat flow problems can be solved through

Use of a Hydraulic Analogue

For several years, many of those active in the air conditioning industry have shown interest in a more exact method for determining heat flow through composite walls under unsteady temperature state conditions. The designer needs to know the peak instantaneous heat load which will be thrown on air conditioning equipment for a specific installation. Mathematical analyses are tedious and time consuming. As such analyses require a working knowledge of higher mathematics, the practicing engineer has resorted to empirical formulas and estimations based on experience.

It is possible to obtain information concerning heat flow by experimental means using thermal heat meters and temperature recorders. However, this is a time-consuming job, since a typical design day must occur during the time these data are being collected, and such tests require either hourly readings or complicated and expensive recording equipment. Further, an actual model of each type of building material would need to be constructed and subjected to climatic conditions.

Basic equations which cover heat flow, fluid flow in the laminar regions and certain direct current circuits are similar. In each case a quantity of fluid, electrical energy or heat may be equated to some variable term multiplied by a constant. The equations shown below illustrate this point.

Quantity = (some constant) (driving force) General Equation
$Q = UA\ (\Delta T)$ Heat Flow

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Member ASHRAE

$Q = K\ (\Delta P)$ Fluid Flow

$Q = \frac{\Delta V}{R}$ Electrical Flow

These equations are valid only for steady state conditions. If the temperature, voltage or pressure differential varies with time, the equations will not be valid and will require that the effect of storage of electrical energy, fluid or heat be taken into account.

As a quicker solution to the problem of periodic heat flow as related to air conditioning, a hydraulic circuit similar to the one shown in Fig. 1 was constructed at the University of Florida. This circuit is based on the similarity of fluid flow to heat flow. A variable head tank indicated surface temperature variations. The various valves were calibrated to represent a resistance to fluid flow which would be equivalent to thermal resistance, and glass tubes were used to represent thermal capacitance; i.e., fluid was stored in these tubes in the same manner that the heat is stored in a wall or roof section of a building. The fluid level in the glass tubes is an indication of temperature at the center of a wall segment. Table I shows equivalent values:

Table I

Heat Flow	Fluid Flow
Btu	Weight of Fluid
Temperature Variation	Head Variation
Thermal Capacitance	Volume of Fluid
Thermal Resistance	Resistance to Fluid Flow
Rate of Heat Flow (Btu/hr)	Rate of Fluid Flow (lb fluid/hr)

Pipe feeding fluid to the weigh tank was held at a constant position and imitated a constant temperature level which would normally exist in an air-conditioned room. By the correct selection of units and using a high viscosity fluid, it is possible to maintain laminar flow and duplicate by simulation the behavior of heat flow through a wall. In order to test the hydraulic circuit a problem was selected which had been solved mathematically and experimentally by the ASHRAE Research Laboratory.

This problem consisted of a pine panel of nominal 2-in. thickness exposed horizontally to variations of sun and air temperatures. The inside surface was in contact with a room temperature of 70 F. A similar wall was set up in the analogue using the same values for thermal resistance, thermal capacitance and inside air film. The variable head tank was made to fluctuate in the same manner as the outside surface temperature of the experimental panel.

The wall was broken down into three segments and the following assumptions were made: (1) that the material was homogeneous, (2) that the capacitance

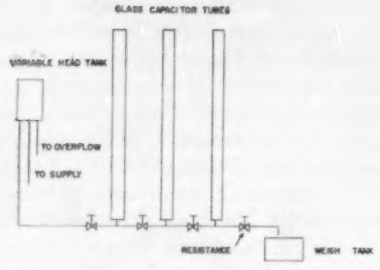

Fig. 1 Hydraulic circuit

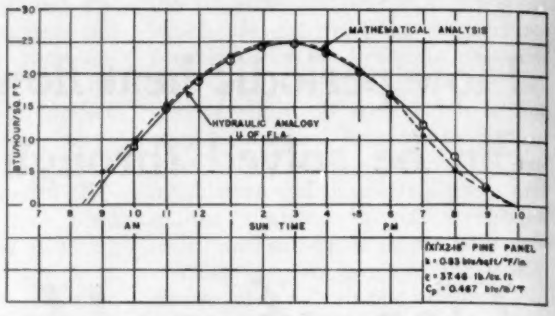

Fig. 2 Comparison of data from mathematical analysis and hydraulic analogue

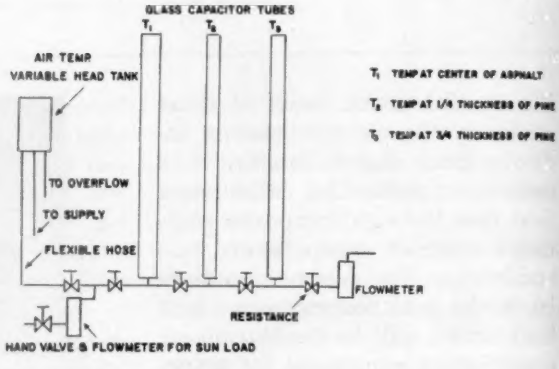

Fig. 3 Expanded hydraulic circuit with additional factors shown

could be lumped at the center of each section and (3) that the thermal resistance would not vary through the temperature range of this experiment.

The second assumption was a variation from a pure mathematical approach to this problem, inasmuch as mathematicians would break the wall into an infinite number of thin sections and sum up these sections between finite limits. This is usually accomplished by means of a series solution. However, the examination of such a solution will show that after the first three parts of a series have been evaluated, the values become small enough to be discarded, so the error introduced by the use of but three wall sections will be insignificant.

All calculations were based on heat flow through one sq ft of roof section. For easy readability it was decided to permit 0.90-in. head variation to be equivalent to 1 F temperature variation. Thus, 100 F of temperature variation could be represented by 90 in. of head variation. Other values must be selected with due regard for the physical properties of the material. In one case, that of a pine panel, 0.0705 lb of fluid was made equivalent to one Btu, and the quantities of fluid stored for each degree temperature rise were equivalent to 0.111 lb. The equipment was calibrated on the basis of steady state conditions and was checked through four tank positions from zero height to maximum height to assure that a linear relationship existed between head tank position and flow.

This was physical evidence that laminar flow existed throughout the system. The head tank was driven by a cam mechanism through a cycle which represented temperature variation of the test panel for a 24-hr period. The time scale was one cam revolution in 24 min or one hr was represented by one min.

Fig. 2 provides a comparison of data obtained by mathematical analysis and hydraulic analogue. Note that the curves are similar and at times coincident. Integrating the area under each curve gives a total quantity of heat flow. This quantity, when compared to the quantity of fluid collected, was in agreement to within 1%.

Tests were conducted for a total of six times on various days, and all test data were found to be in substantial agreement. Since the hydraulic analogue solved successfully problems involving periodic heat flow where surface temperature varied, the circuit was expanded as shown in Fig. 3 and the following additions were made.

The weight tank was replaced by a flow meter which gave an instantaneous reading for fluid flow leaving the hydraulic system. Oil from a supply tank was forced under pressure through a flow meter and injected into the circuit between a valve representing the outside air film and a valve representing the first segment of the wall. This flow was metered into the system in such a manner as to represent the radiant sun load impinging on the exposed segment.

Re-radiation from the roof's surface was accounted for by reducing the radiant sun load by an amount equal to the energy loss due to re-radiation. To obtain the energy loss an emissivity factor was chosen which was suitable for the roof under test. The air temperature was represented by a variation in position of the head tank.

It was decided to set up a problem involving a section of roof material identical to the one used in a previous experiment at the University of Florida involving the cooling of roofs by water spray. This experiment is covered in "Roof Spray for Reduction in Transmitted Solar Radiation."* A cross section of this roof is shown in Fig. 4. This was constructed of nominal size 2-in. pine boards covered with a layer of builder's felt, upon which was poured sufficient asphalt, which was then covered with a layer of ¼-in. gravel to give a thickness of ½ in. Handbook values were used for thermal resistance of the material, thermal capacitance and density.

An outside air film was calculated, using climatic data obtained from Bulletin No. 5, "Climatic Data for the Design and Operation of

* Journal Section, Heating, Piping & Air Conditioning, September, 1950.

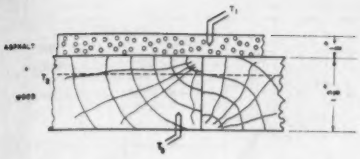

Fig. 4 Cross section of roof material used

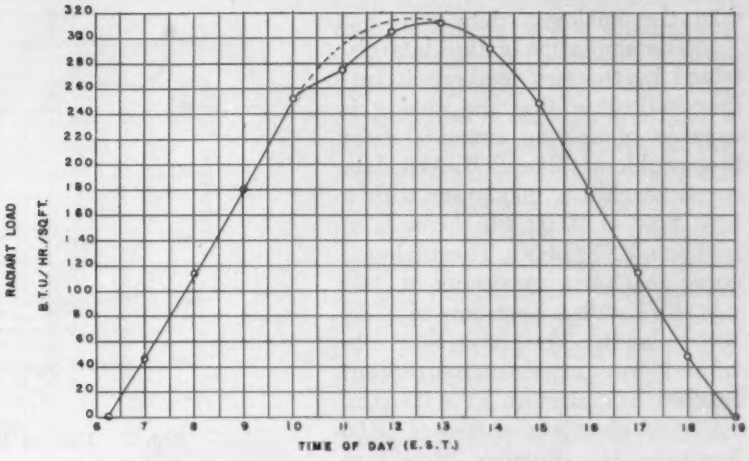

Fig. 5 Typical design solar radiation curve

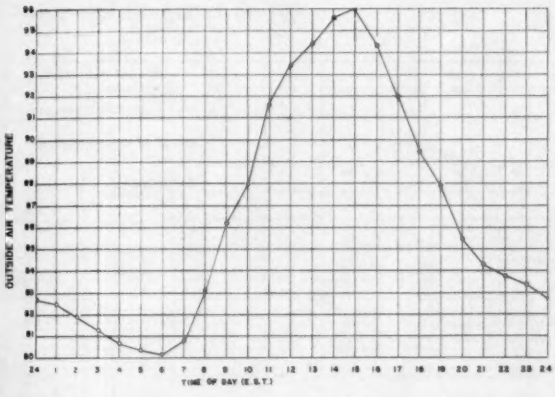

Fig. 6 Daily air temperature variation plotted against time

Air Conditioning Systems in Florida" (Engineering and Industrial Experiment Station, May, 1939). The inside air film used was based on information taken from the ASHRAE GUIDE. A cam was fabricated to drive the head tank in the same manner that a typical design day's temperature might vary. Data for the fabrication of this cam were taken from Bulletin No. 5, mentioned previously.

A study was made of radiant energy curves which have been collected on a daily basis over the past seven years at the University of Florida. On the basis of this study, a typical design solar radiation curve was developed. This curve is shown in Fig. 5. Fig. 6 shows a typical daily air temperature variation plotted against time. The roof as shown in Fig. 1 was divided into three segments. The first segment represented the gravel and asphalt roof covering. The last two segments represented one-half of the roof section, respectively. In this case it was necessary to make up an insert for the tube used to represent the asphalt and gravel in order to reduce the volume of this tube to match the heat storage of ½-in. asphalt and gravel.

The roof as described previously was set up in the analogue and calibrated under steady state conditions and was checked through a complete cycle of expected temperature variations to ascertain that laminar flow was in existence throughout this entire range. The head tank was cycled using the cam fabricated from the temperature gradient shown in Fig. 6 and was found to be positioning the tank properly with respect to time. Equivalent sun loadings were impressed into the hydraulic system each 15 sec, thus giving the effect of a step function rather than a continuous input.

In order that ambient temperature conditions might not affect the viscosity of the fluid in the analogue, the room in which it was located was air conditioned and maintained at 80 F. Both flow meters were calibrated, using a stop watch and a weigh tank. A constant fluid temperature of 80 F was maintained and anytime the temperature change in the fluid exceeded 1 F, the experiment was stopped while the fluid temperature returned to 80 F. The analogue was put in operation with the weigh tank in the position which represented the temperature level at 5:00 a.m. At 6:15 a.m. radiant energy was fed into the hydraulic system, and the machine was permitted to cycle for one complete revolution of the driving cam (24 min), which represented one day's operation, before readings were taken. Fig. 7 gives a plot of temperature variations at a point three-fourths the distance from the top of the pine roof section as compared to time of day.

Fig. 8 provides a chart of actual heat flow through a segment of roof identical to the one set up in the hydraulic analogue, this being made from data taken using heat flow meters on a section of roof which was exposed to radiant sun loading and outside air temperature during a typical Florida summer day. Further information covering the method of obtaining the actual heat flow data is contained in the article, "Roof Spray for Reduction in Transmitted Solar Radiation." Also shown on Fig. 8 is a plot of predicted heat flow obtained by means of the hydraulic analogue and a predicted heat flow curve calculated by means of the SOL-AIR temperature method, this method of heat flow calculation having been developed by ASHRAE. Data obtained by means of the hydraulic analogue and actual test are almost identical with respect to magnitude, but the actual data show a peak flow 1 hr later

than the analogue data.

The similarity of the data obtained by the two methods is further indication that the analog is capable of solving practical heat flow problems. The SOL-AIR data curve reaches a maximum with a heat flow of 20.00 Btu/hr/sq ft at 2:30 p.m. EST (14:30). The analogue curve reaches a maximum at 3:00 p.m. EST with a heat flow of 17.50 Btu/hr/sq ft. The heat flow obtained by actual measurement reaches a maximum at 4:00 p.m. EST and gives a heat flow of 18.00 Btu/hr/sq ft. Further data are needed before a comparison can be made of the ability of the analogue (and the SOL-AIR method) to predict heat flow.

In an effort to test the machine further, a thermocouple was embedded one-fourth of the distance from the ceiling side of the roof (three-fourths the distance from the top of the roof as shown in Fig. 4). This is the same point used in the analogue set-up, and temperatures recorded here should be in agreement with those predicted by the analogue. Fig. 7 shows a plot of predicted temperatures, obtained by the hydraulic analogue, at a point in the roof section for a 24-hr period. Also shown on Fig. 7 is a plot of actual temperature measurements taken at the same point, on August 10, 1955, in an existing roof structure. The plots are in fair agreement until approximately 2:00 p.m., at which time a temperature difference of four deg is noted, with the actual temperature being higher. No ready explanation can be given for this discrepancy unless a change in the air film coefficients could account for it.

A study of the solar radiation curve for this same day indicates that the solar load on the actual roof section was quite close to the solar design curve used in the analogue. A study of the air (outside) temperature curve for that day also indicated that the outside air temperature was quite close to the design conditions which were used in the analogue. Since this is not a major discrepancy, it would not change the over-all heat flow to any great extent.

The fact that the actual temperature curve falls off quite rapidly after 5:30 p.m. and reaches a lower temperature level than that

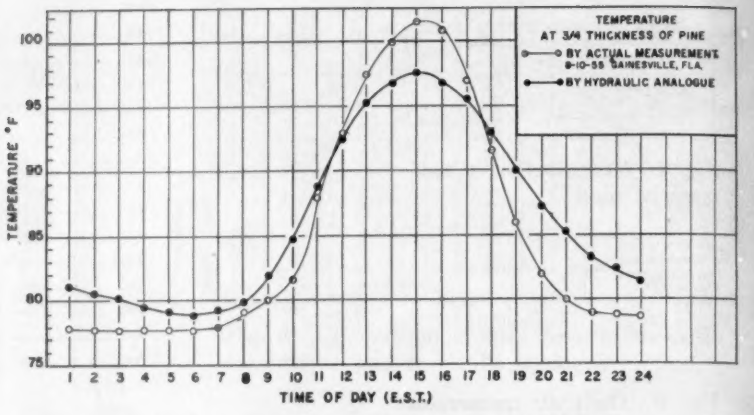

Fig. 7 Plot of temperature variations at ¾ the distance from roof section as compared to time of day

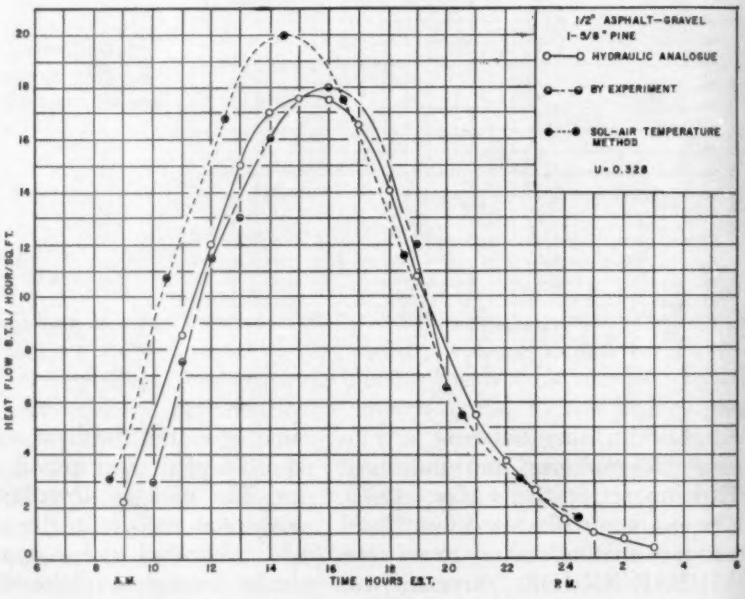

Fig. 8 Actual heat flow through roof segment

predicted by the analogue can be explained by the fact that several tall pines cast a shadow on the test panel from 3:45 p.m. until darkness; thus, the panel was not subjected to the full sun load expected nor in the amount calculated for the analogue. The same lag was noted in the time it took the shadow on the roof section to lower the temperature at a point down in the roof panel, existing between the maximum sun loading on the roof surface and the maximum temperature of a point in the roof section. This time lag is approximately one hr, 45 min.

Since the analogue was capable of solving complex heat transfer problems, the following panels were set up in the analogue and subjected to the typical design conditions for temperature and solar load as previously discussed in this paper:

(1) 2-in. concrete with ½-in. asphalt and gravel on top of panel
(2) 2-in. concrete with 1-in. of cork board between top of panel and ½-in. asphalt and gravel
(3) 2-in. concrete with ½-in. asphalt and gravel on top and 1-in. cork board on bottom of panel

The reasons for choosing the above types of roof structure for study were threefold: (1) since these types of panels are used widely in building construction, further information concerning heat flow through such panels would be useful to practicing engineers; (2) a

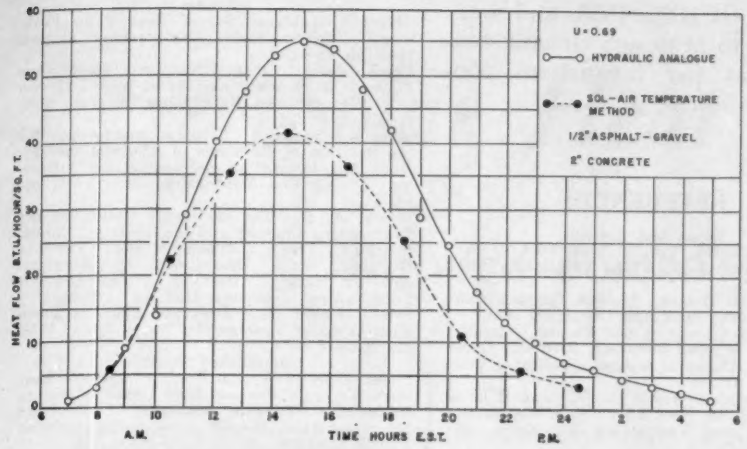

Fig. 9 Heat transfer against time using uninsulated concrete with asphalt gravel on top

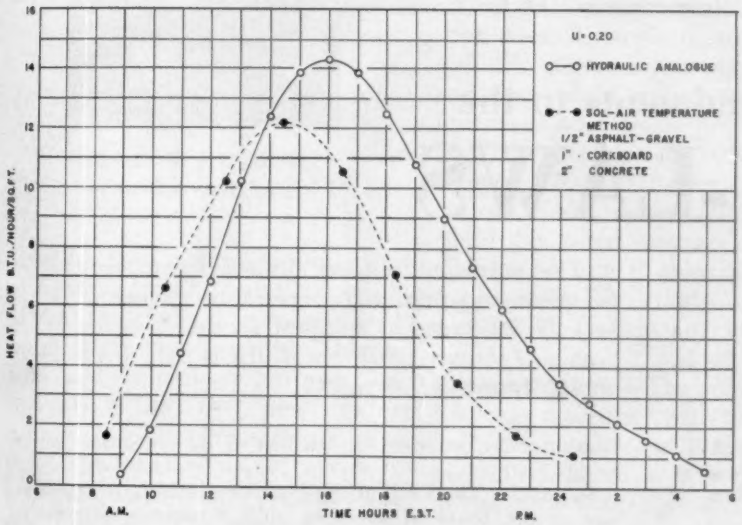

Fig. 10 Heat transfer against time using concrete and cork with asphalt gravel on top

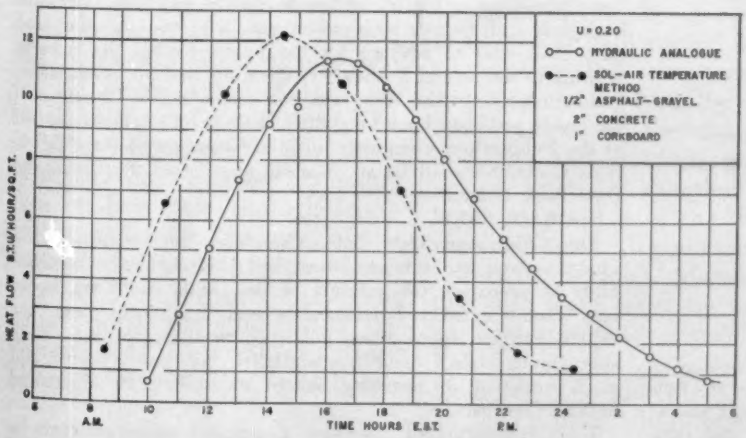

Fig. 11 Heat transfer against time using concrete slab roof with asphalt gravel on top and cork on the bottom

comparison of the heat flow through roof structures of the cited types would point up the need for insulation and would also show that the location of insulation within a complex wall has an effect on the heat transmitted through the wall; (3) an additional comparison could be made between the analogue method of determining heat transfer and the SOL-AIR temperature method of determining heat transfer. Results of these tests are shown in Figs. 9, 10 and 11.

Fig. 9 shows a plot of heat transfer against time for an uninsulated two-in. concrete slab with ½ in. of asphalt gravel on top. The analogue predicts a peak heat flow of 55 Btu/hr/sq ft at 3:00 p.m. EST (15:00), whereas the SOL-AIR method predicts a peak heat flow of 42 Btu/hr/sq ft at 2:30 p.m. EST (14:30).

Fig. 10 is a plot of heat transfer against time for a two-in. concrete slab type roof with one-in. cork board on top of the concrete and ½-in. asphalt and gravel poured on top of the cork board. The analogue predicts a peak heat transfer of 14.3 Btu/hr/sq ft of roof surface and predicts that this peak will occur at 4:00 p.m. EST (16:00). The SOL-AIR temperature method predicts a peak heat flow of 12.2 Btu/hr/sq ft and predicts that this peak will occur at 2:30 p.m. EST (14:30).

Fig. 11 shows a plot of heat transfer against time for a two-in. concrete slab roof with ½-in. asphalt gravel on top of the roof and a one-in. cork board on the bottom of the roof panel. In this case the peak heat flow of 11.4 Btu/hr/sq ft of roof surface was predicted by the hydraulic analogue and this peak was to occur at 4:30 p.m. EST (16:30). The SOL-AIR temperature curve plotted on Fig. 11 is identical to the SOL-AIR temperature curve plotted in Fig. 10, since this method has no means of taking into account the position of insulation with respect to a roof section or, for that matter, any type of building construction.

A study of results obtained by the analogue shows that the addition of one-in. cork board, when placed between the asphalt gravel coating and the roof structure, reduces the peak instantaneous heat gain from 55 Btu/hr/sq ft to 14.3

Btu/hr/sq ft and increases the time lag for this peak by one hr. By placing the insulation on the bottom of the concrete slab, a further reduction of heat flow was predicted, dropping the total flow to 11.4 Btu/hr/sq ft of roof surface and increasing the time lag by 30 min. Thus, not only is insulation in itself important, but it would appear from these tests that the location of insulation is important.

Values used in the SOL-AIR temperature method were taken from Table IX, Chapter 13 of the ASHAE GUIDE, 1958, and were adjusted to fit design climatic conditions as per instructions contained therein.

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AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR CONDITIONING ENGINEERS

Amendments to the BY-LAWS

As read to the membership at the Semiannual Meeting held in Chicago, February 13-16, 1961 and voted upon affirmatively at the 68th Annual Meeting in Denver, Colo., June 26-28, 1961.

The numbered sections printed herewith supplant those originally provided as of January 29, 1959, and as amended previously by vote of February, 1961. See ASHRAE JOURNAL, April, 1961, page 79.

Sec. 3.1 The grades of membership in the Society shall be designated as follows: (A) Honorary Members, (B) Presidential Members, (C) Fellows, (D) Life Members, (E) Members, (F) Associate Members, (G) Affiliates, and (H) Students.

Sec. 3.4 Renumber as Section 3.5 Life Member—A Member who has been a Member in good standing for thirty (30) years; and who has attained the age of sixty-five (65) years. He shall retain all the rights and privileges of his former membership grade. He shall not be required to pay any annual dues or any other fees.

Sec. 3.5 Renumber as Section 3.4.

Sec. 8.7 The names of the General Committees of the Society shall be as follows:

Executive	Education
Finance	Honors and Awards
General & Administrative	International Relation
Coordinating	
Technical Coordinating	Meetings Arrangements
Divisional Advisory	Membership Development
Regions Central	Nominating
Chapters Regional	Professional Development
Long-Range Planning	Program
Advisory Board—Past Presidents	Publications
Admissions and Advancement	Public Relations
Charter & By-Laws	Research & Technical
Exposition	Standards
	Guide & Data Book

Sec. 8.84 The Technical Coordinating Committee shall consist of the Chairmen and Vice Chairmen of the Program, Publications, Research and Technical, Standards and Guide and Data Book Committees, and shall have a Vice-President as Chairman. In addition, there shall be three (3) Members-at-Large appointed by the President of the Society. It shall be the duty of the Committee to coordinate the activities of the committees named in this Section.

Sec. 8.8.21 The Publications Committee, subject to the direction of the Board of Directors, shall formulate the editorial policies of the Society and all of its publications. The chairman may appoint sub-committees of one (1) or more members to review and report to the committee on the quality and appropriateness for publication of papers and bulletins intended for presentation or presented at Society meetings and the discussions thereof. In the performance of its functions the said committee and its sub-committes shall be subject to the following conditions: (A) that the data recommended for publication shall tend toward the professional education of the individual engineer; (B) that such data shall be free from commercial bias; and (C) that such data shall tend to advance for the public benefit the sciences relating to the arts of heating, refrigeration, and air conditioning and ventilation, or the allied arts and sciences. The Chairman of the Guide and Data Book Committee shall be an ex-officio member of the Publications Committee and the Chairman of the Publications Committee shall be an ex-officio member of the Guide and Data Book Committee.

Sec. 8.8.25 NEW

The Guide and Data Book Committee shall consist of a Chairman and two subcommittees, and shall have the responsibility of preparing two volumes of the Guide and Data Book: one volume entitled Fundamentals and Equipment, and one volume entitled Applications, published in alternate years. Each subcommittee shall have responsibility for one volume and shall consist of six members, one of whom shall be designated as Vice-Chairman.

Upon completion of a volume a new subcommittee shall be appointed to prepare the next revision of that volume. At least three (3) members of the new subcommittee shall not have served on the previous subcommittee for that volume.

Solar Heat Gains

through domed skylights

L. F. SCHUTRUM
Member ASHRAE

N. OZISIK

Solar heat gains through four types of light controlling and three types of flat glass skylights were reported upon some time ago by Messrs. Vild and Parmelee[1]; however, information on the heat gains through a quite common type skylight, the plastics dome, which is becoming increasingly popular, was not available. The purpose of this study was to obtain data so that solar heat gains through acrylic plastics dome skylights could be determined by the air-conditioning engineer. Three domes having high, medium and low light transmitting qualities were selected to cover the range of normal practice, and a fourth dome which was painted black was tested in the ASHRAE Solar Calorimeter.[2]

This calorimeter has been used over the years for compiling data on heat flow through fenestration under the guidance of the Society's Technical Advisory Committees. The present study was started under the guidance of the former TAC on Heat Flow Through Fenestration, which has been discontinued through reorganization of the Technical Committees.

L. F. Schutrum is a Research Supervisor, ASHRAE Research Laboratory. N. Ozisik is with the Oak Ridge National Laboratory. This paper was presented at the ASHRAE 68th Annual Meeting in Denver, Colo., June 26-28, 1961.

TEST APPARATUS

The heat gain through the skylights was determined by means of the solar calorimeter, from the rate of flow of the circulating mixture of ethylene glycol and water, and the difference between inlet and outlet fluid temperatures. Adjustments were made for the heat flow through the back of the calorimeter. The calorimeter could be rotated to any desired azimuth and could be tilted through 90 deg about a horizontal axis. The opening to which the skylight was fitted was 44¼ x 44¼ in.

Instrumentation – The solar radiation was measured with two Eppley pyrheliometers mounted on the face of the panel on opposite sides of the calorimeter opening. A third pyrheliometer was used at times to measure the direct normal solar radiation. Other weather conditions recorded were of the dry bulb and wet bulb temperatures, wind velocity and direction, and an indication of the cloudyness and hazyness of the sky. A convection compensated radiometer[3] was used to measure the low temperature radiation coming from the sky. The temperatures of the calorimeter, skylight and outdoor air were sensed by thermocouples, and as much as possible of the data was recorded on a 16-point electronic roll chart recorder. Manual readings also were taken by means of an electronic potentiometer.

Skylights–The four plastics domes and one ceiling light diffuser which were used in these tests were made from 4 by 4 ft sheets of acrylic plastics to fit the opening of the calorimeter (44¼ x 44¼ in.). Three domes, and the diffuser being held by the technician, are shown in Fig. 1. The dome height at the center was approximately ¼ of the span, and the height of the light diffuser was about 2 in. Two of the domes shown in the picture are light-diffusing types, and the third was painted black for special test purposes. A fourth dome, which is transparent, is shown in Fig. 2, in position on top of the calorimeter ready for testing. In this case it is mounted on a special curbing, 18 in. high, but in other tests a 9-in. curb or no curbing at all was used. Also seen in Fig. 2 are two pyrheliometers and the convection compensated radiometer. A brief description of the domes is given in Table I.

Test Procedure – The solar calorimeter was set in a horizontal position for most of the tests except where special angles of incidence were desired, in which case the calorimeter was tilted. Tests were made with the calorimeter in a fixed position or with the calorimeter adjusted to follow the sun. Data were gathered throughout the 10 or 15-min test periods, on

Fig. 1 Plastics domes and light diffuser

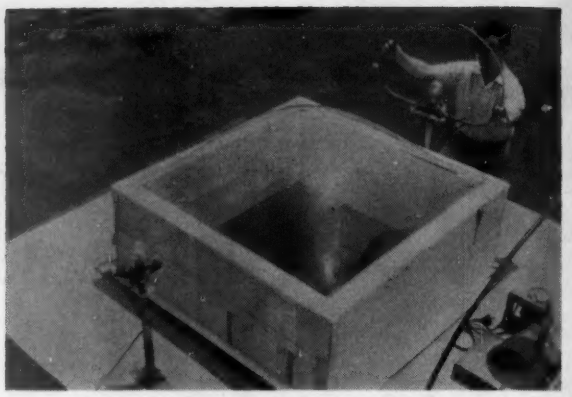

Fig. 2 Transparent dome and 18-in. curbing mounted on solar calorimeter for testing

the solar radiation, desired angles, outdoor environment, calorimeter liquid flow rate and temperature rise, and the various surface temperatures.

Analysis—Solar energy falling upon a dome skylight will be reflected partly from the dome, partly absorbed, and the remaining radiation will pass on through into the interior of the building. The reflected radiation is rejected and is not involved in the heat balance. Because the skylight is heated by the absorbed radiation, it will increase in temperature until the rate of solar radiation absorbed equals the rate of heat loss by radiation and convection to the outdoors and indoors, and by conduction heat flow to the supporting members. The radiation and convection and the aforementioned solar radiation passing through the skylight are the components which constitute the total solar heat gain.

METHOD OF ATTACK

At the outset, it was decided to compute the solar properties of the domes to be tested from measured properties of the flat plastics sheets from which the domes were formed. These theoretical properties of the domes could then be verified by experimentation in the solar calorimeter and re-evaluated if necessary. Convection and radiation exchanges between the skylight and interior would be determined by test.

Transmittance—The solar transmittance is defined as the ratio of the rate of solar energy passing through a material and the incident solar energy. Reflectance similarly is a ratio of reflected to incident radiation. The solar transmittances and reflectances of flat sheets of three thicknesses and of the three basic types of plastics were measured by means of an Eppley pyrheliometer. These measurements were made for angles of incidence up to 60 deg. The transmittances and reflectances of flat sheets for various angles of incident solar radiation are given in Appendix A.

From the transmittances and reflectances of the flat sheets, the solar transmittances and reflectances were computed for a hemispherical shaped dome and a spherical segment (Fig. 3). The method of computation is described in the appendix, but essentially these properties were computed by dividing the dome into zones, having a limited range of incidence angles, and using the transmittances and reflectances determined from the flat sheets for the average incident angle of each zone. For the transparent dome, the transmitted radiation and radiation reflected from the outside of the dome were assumed to be specular. After the energy had entered through the dome any reflections from the dome surfaces were considered diffuse. For the diffusing domes both transmittances and reflectances were considered diffuse in nature.

Dome Skylights (No Curbing)—The computed transmittances of three types of flat plastics sheets, hemispherical domes and spherical segments are shown by the curves of Fig. 3. Note first of all that the transmittance is based on the normal intensity of the direct solar radiation instead of the incident radiation. This relationship was used so that the transmittance at a 90-deg incidence angle could be finite instead of being infinite (resulting from the incident radiation on a horizontal plane being zero). Experimental points determined from calorimeter tests of actual domes made of three types of plastics also are plotted in Fig. 3.

The calorimeter measured the total heat transfer which occurred through the domes. By subtracting the convection-radiation heat exchange between the dome and the calorimeter, and the diffuse radiation component, the transmittances for direct solar radiation were ob-

Table I—Specific Features of Domes

Dome Designation	Solar Transmittance of Flat Sheet (Direct Normal)	Plastics[a]	Thickness of Base Sheet in.	Dome Height in.[b]
I Clear (Transparent)	0.86	Acrylic Colorless	3/16	11¼
II Medium Transmittance (Translucent)	0.52	Acrylic White W-2447	3/16	11¼
III Low Transmittance (Translucent)	0.27	Acrylic White W-7328	3/16	11¼
Light Diffuser (Translucent)	0.58	Acrylic White W-2447	1/8	2⅓

[b] All domes 44¼ x 44¼ in. at the base
[a] Number designation by Rohm & Haas Company

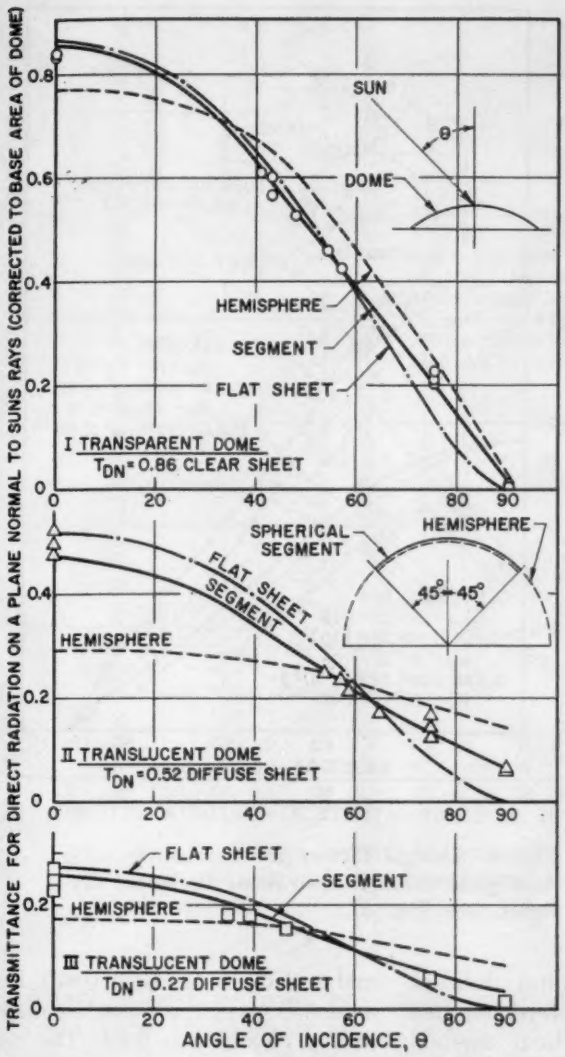

Fig. 3 Transmittance of domes for solar radiation

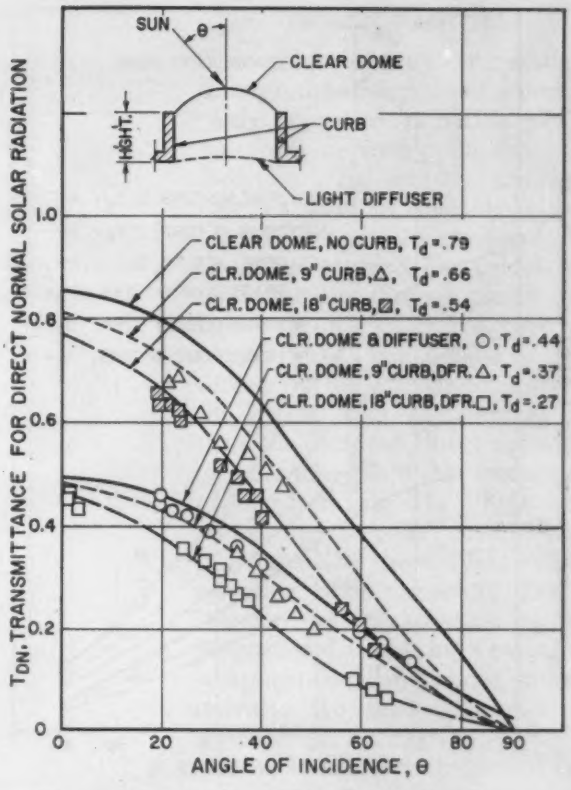
SUN
θ
CLEAR DOME
CURB
HGHT.
LIGHT DIFFUSER
CLEAR DOME, NO CURB, T_d = .79
CLR. DOME, 9" CURB, △, T_d = .66
CLR. DOME, 18" CURB, ☐, T_d = .54
CLR. DOME & DIFFUSER, ○, T_d = .44
CLR. DOME, 9" CURB, DFR. △, T_d = .37
CLR. DOME, 18" CURB, DFR. □, T_d = .27
T_{DN}, TRANSMITTANCE FOR DIRECT NORMAL SOLAR RADIATION
ANGLE OF INCIDENCE, θ

Fig. 4 Transmittance of skylight, clear dome

tained. These convection-radiation and diffuse radiation corrections are a small part of the total gain for near-normal incidence angles, but become a major part of the total gain at high incidence angles where the direct component is quite small. It will be noted in Fig. 3, that the experimental points correlate best with the curve for the segment (height to diam = 0.21) which corresponds closely with the shape of the actual domes (height to width = 0.25). Tests also were conducted with the calorimeter to determine the effect of the solar azimuth on the transmittances of the domes. The transmittances were found to be practically independent of the solar azimuth, i.e whether the sun's rays were in a plane parallel to the sides of the dome, or parallel to the diagonal.

EFFECT OF CURBINGS AND LIGHT DIFFUSER

To investigate the effect of curbing height on solar heat gains, two curbs, each 9 in. high, were constructed. These curbings were used to give curb heights of 9 in. and combined to give 18 in. The corresponding ratios of width to height are approximately 5 and 2½. The curbings were well insulated to reduce heat flow through the sides to a minimum and instrumented so that this heat leakage could be measured. The inside surfaces were painted with three coats of paint of known light reflectance (0.75). The solar reflectance of the walls of the curbing as measured with a pyrheliometer was 0.7 as compared to magnesium carbonate (0.95).

The solar transmittances of skylights of various combinations of dome, curb and the light-diffuser were computed from the known solar properties of the components (90-deg segment properties for the dome) by means of the transfer-function method given in reference 4 as described in the appendix. At the same time, tests were conducted with the solar calorimeter to substantiate the calculated transmittances, and to evaluate the factors effecting the convection and radiation exchange between the skylight and the calorimeter.

The solar transmittance for direct normal solar intensities, for a clear dome with and without the light diffuser, and for curbing heights of zero, 9, and 18 in., are given in Fig. 4. The points in the figure were obtained from the tests with the solar calorimeter. The curves shown are theoretical values which have been adjusted to fit the data better.

It was observed that some of the solar radiation passing through the edges of the clear dome caused high intensity bands of light to fall in the curbing. In order to adjust the theoretical calculations for this effect, some of the transmitted energy was assumed to be diffuse. The amount chosen was 25% of the total transmittance, which improved the correlation between observed and calculated as given in Fig. 4.

The amount of solar energy passing through the transparent dome and into the interior of the calorimeter without first falling upon the curb depends also, of

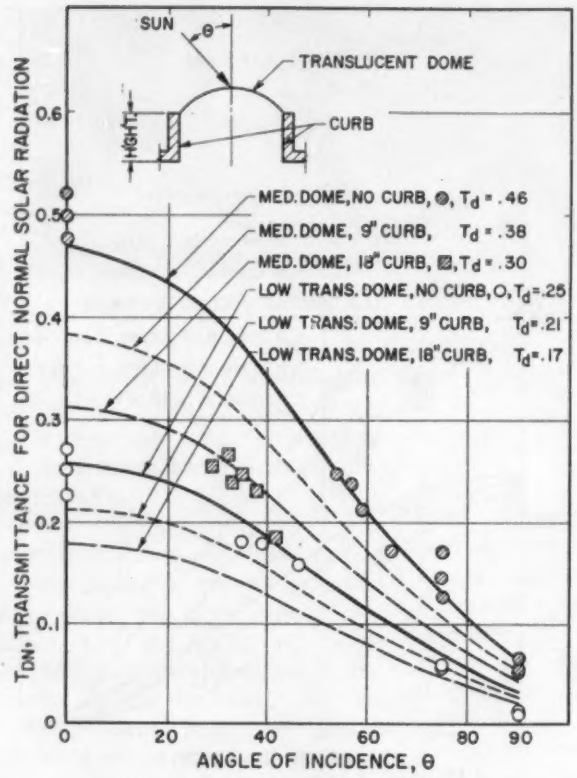

Fig. 5 Transmittance of skylights, translucent domes

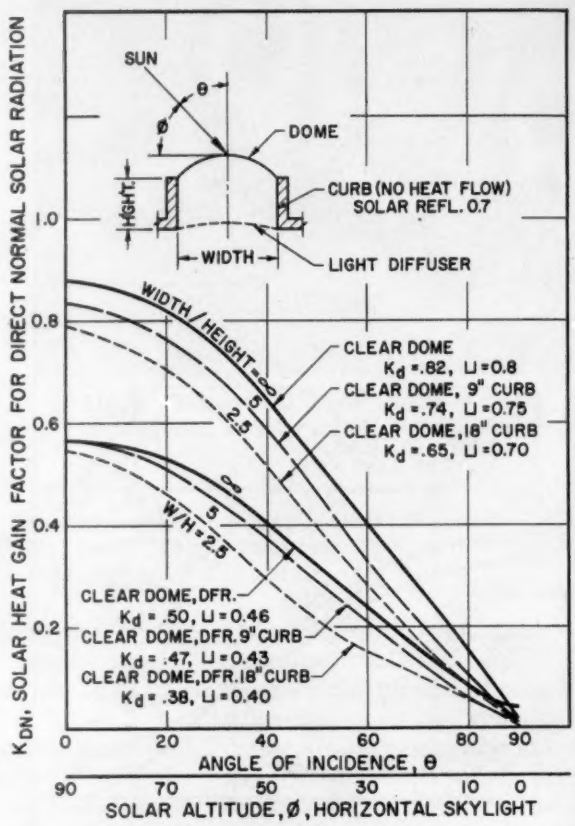

Fig. 6 Design factors for estimating solar heat gains through clear-dome horizonal skylights (see Eq. 3)

course, upon the solar azimuth, because the curb in these tests was square in shape. The solar transmittances of the skylights given in the figure are for zero solar azimuth (sun in plane perpendicular to side). For the solar azimuths of 45 deg (diagonal), the transmittances are lower for angles of incidence other than 90 and zero deg. For a 45-deg incidence angle, and 18-in. curbing, the reduction is about 10%, and for 9-in. curbing, somewhat less.

The transmittances for diffuse or sky radiation also are given. These were computed from the curves of Fig. 4 for a uniform sky radiation, however, the actual sky radiation distribution is non-uniform and varying. Pyrheliometer and calorimeter tests indicate that the transmittance for diffuse radiation should be about 10% lower than those reported in Fig. 4 with the exception of the clear dome alone. Had the lower diffuse transmittance value been used, the experimental points in general would be slightly higher.

The theoretical transmittance curves of the skylights in which diffusing-type domes were used are shown in Fig. 5. The test points for the medium transmittance dome and 18-in. curbing, and data of Fig. 3 without curbs, were assumed to verify the theoretical method (Appendix B). The skylights with curbing give transmittances which are directly proportional to the transmittance of the dome used alone. This is the result of assuming the domes transmit diffusely and that the curbing is also a diffuse reflector.

CONVECTION RADIATION GAIN

The convection and radiation heat exchange coefficients between the solar calorimeter and the skylight were determined by measurement without solar radiation entering the calorimeter. This was done by testing during the day with a black dome, which was opaque to solar radiation, or with the other transparent or translucent domes at night.

The convection heat exchange was determined by subtraction of the computed radiation exchange between skylight and calorimeter from the total calorimeter heat flow. A hemispherical emissivity of 0.83 was used for these calculations, which corresponds to a normal emissivity of about 0.875. Reference 6 gives the normal emissivity for Plexiglas as 0.89. The convection heat transfer downward from the dome surface to the calorimeter was quite small, and under the conditions for a horizontal skylight the combined radiation-convection coefficient was about unity.

With the convection heat flow upwards from calorimeter-to-dome, the convection coefficient was of the order of 0.7 to 0.9 Btu/hr/sq ft/F, which gives a combined coefficient of about 1.4-1.9 Btu/hr/sq ft/F.

The heat exchange between the skylight dome and the outdoor, takes place by convection exchange with the outside air and low temperature radiation exchange with the sky and the surroundings which are seen by the dome. The radiant energy emission from cloudless skies can be estimated from the information given in reference 3 and was measured during many of the tests by a low temperature radiometer which is described in the same reference. A number of attempts were made to determine

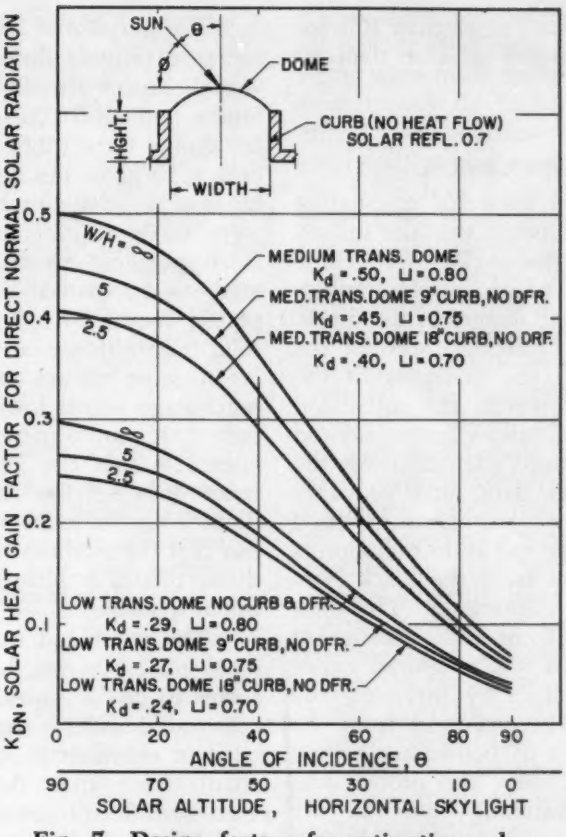

Fig. 7 Design factors for estimating solar heat gains through translucent-dome horizontal skylights

the convection heat exchange between the dome and outdoor air, by measuring the total loss from the dome at night by means of the calorimeter, and the radiant energy loss to the sky. The results were inconclusive.

Knowledge of the solar energy absorbed by the domes must be had in order that the portion of this energy which enters the room via convection and radiation can be determined. This absorptance was determined from the relationship of transmittance + reflectance + absorptance equals unity, since the first two solar properties were already determined and transmittance verified by experimentation.

The portion of the solar energy absorbed by a single-dome skylight (no curbings) which enters the space B is

$$C + R = U \frac{\alpha_{DM} I}{h_o} \qquad (1)$$

and the heat entering due to a temperature difference between indoor and outdoor air is

$$C + R = U (t_o - t_i) \qquad (2)$$

These equations can be combined if the appropriate U-value is used. The U-value does vary somewhat with the combined magnitudes obtained from equations 1 and 2.

$$C + R = U \left[\frac{\alpha_{DM} I}{h_o} + (t_o - t_i) \right] \qquad (3)$$

In a similar manner the convection and radiation gain by the room from the skylight was related to the solar energy absorbed by the dome, curbing, and diffuser, where used, with the simplifying assumption that no heat flows through the curbing, and that thermal storage effects could be neglected.

This method of solution is given in Appendixes D and E. For design heat gain calculations, the over-all outside coefficient of heat transfer was taken as 4 Btu/hr/sq ft/F. With this coefficient the calculated convection-radiation gain to the room was comparable with the test data. A high degree of accuracy cannot be expected in these data because the heat flow through the curbing was neglected in order that the results would not be too complex for practical application.

DESIGN FACTORS

The solar energy transmitted through the skylight, and a portion of the convection-radiation gain to the room depend upon the solar radiation. To simplify the heat gain calculations solar heat gain factors K_D and K_d were developed in earlier papers.[9] For calculating the heat gains through plastic skylights, the K_D factor has been replaced by K_{DN} which must be multiplied by the normal intensity of the direct solar radiation rather than the incident direct radiation to determine the direct component of solar heat gain. The diffuse component may be determined by multiplying K_d by the incident diffuse radiation. The remaining part of the convection-radiation gain is proportional to the indoor-outdoor temperature difference; consequently, the total gain can be given as:

$$\text{Total gain} = K_{DN} I_{DN} + K_d I_d + U (t_o - t_i) \qquad (4)$$

These K factors and U-values suggested for estimating solar heat gains are given in Figs. 6 and 7.

Use of Design Factors – The K values are intended to give the air-conditioning engineer a means of estimating the solar heat gains through these plastics skylights for almost any location. Suppose one wishes to know the solar heat gain through a horizontal skylight having a clear dome and light diffuser with a ratio of width to height of about 5. The performance curve of this skylight is given in Fig. 6 (second curve from bottom). For 11 a.m., Aug. 1, 40 deg North Latitude, the 1960 GUIDE, Chapter 13, Table 6, gives the solar altitude as 64.5 deg. The direct normal radiation is about 286 Btu/hr/sq ft and the diffuse radiation falling on a horizontal surface is about 34.5. These values are from Table 4, Chapter 13, of the GUIDE. From Fig. 6 of this paper, the K_{DN} value from the curve is 0.475, K_d is 0.47 and U is 0.43. Let $t_o - t_i$ = zero. From equation 4

$$\text{Solar gain} = 0.475 \times 286 + 0.47 \times 34.5 + 0.43\,(o) = 136 + 16.2 = 152.2 \text{ Btu/hr/sq ft of opening}$$

The solar heat gains computed by

this method are given in Table II for three latitudes and design conditions for August 1, as given in the GUIDE.

Total heat gain is obtained by adding the product of indoor-outdoor temperature difference and the U-value to the Table II solar gains. In our example the total heat gain for an outdoor air temperature of 87 F and indoor at 80 F is

$$\text{Total gain} = 152.2 + 0.43\,(87 - 80) = 155.2 \text{ Btu/hr/sq ft}$$

The solar heat gains reported in Table II are based on a solar reflectance of 0.7 for the curb. Theoretical calculations show that for a curb having a reflectance of 0.4 the transmittance of the skylight is reduced, but more solar energy is absorbed by the curb and, consequently, the convection-radiation component of the heat gain is increased. These two factors counteract each other, and for the range of curb reflectance 0.4 to 0.7, the total solar heat gains are essentially unchanged. The exception to this is for a skylight consisting of a clear dome and curb which deviated more than 10% for incidence angles greater than 60 deg.

DISCUSSION

The method used for calculating the solar transmittance and reflectance of the domes does not include the energy which is lost through the edges and flanges of the dome, nor does it take into account the variation in the thickness of the plastics. However, the calculated solar transmittances are considered to be confirmed adequately by the experimental data, and probably represent the maximum transmitted energy, since but little radiation is reflected out from the black interior of the calorimeter. The solar reflectance of one of the translucent white domes was measured under natural sunlight by surveying the solar radiation reflected from the dome with a pyrheliometer having a restricted view. This process was so time-consuming that the sky conditions varied considerably during the test; and, consequently, only approximate reflectance values were obtained.

The purpose of skylights is, of course, to provide illumination. One would expect the light transmittances and solar transmittances of the domes to be quite similar. Reference 10 gives the light transmittance of three domes, two of which were made from material of the same specifications as the low and medium transmittance domes reported here. By comparison the solar transmittance curves are similar in shape but are about 10 to 15 percentage points lower in magnitude. Also to be noted in this reference is that the light transmittance for the low transmittance dome is shown to be nearly diffuse but that the medium transmittance dome retains a little of its directional qualities.

The combined outside coefficient used for the design factors seems to fit the experimental data reasonably well. Variations of the outdoor coefficient have but a small effect upon the total heat gain for the dome with curbing and diffuser, and a somewhat larger effect upon heat gains for the dome alone. It has no effect upon the transmitted portion but only the

Table II—Solar Heat Gains through Horizontal Domed Plastics Skylights
August 1, Clear Atmosphere[a] (44 1/4 by 44 1/4-in. opening)
Btu/(hr) (sq. ft of opening)

Skylight			Clear Dome & Light Diffuser			Clear Dome (No Diffuser)			Med. Trans. Dome (No Dfr)			Low Trans Dome (No Dfr)			Flat Glass Reg. Plate[e]
Curb Height,[b, c] in.			0	9	18	0	9	18	0	9	18	0	9	18	0
Ratio	Width[d] / Height		∞	5	2½	∞	5	2½	∞	5	2½	∞	5	2½	∞
Latitude	Suntime														
	AM														
30 deg North	6	6	15	12	11	26	20	17	19	17	15	11	10	9	16
	7	5	47	40	30	80	63	54	48	43	38	27	25	23	64
	8	4	80	73	54	136	116	101	77	70	62	44	40	37	124
	9	3	118	112	85	192	174	150	109	98	88	63	57	53	178
	10	2	152	143	122	238	220	198	137	123	111	79	73	66	220
	11	1	172	167	150	266	249	229	153	137	124	89	82	75	246
	12	PM	178	175	161	278	260	240	159	142	129	93	84	78	255
	AM														
40 deg North	6	6	20	17	14	36	27	23	25	22	20	14	13	12	23
	7	5	50	43	34	85	68	58	51	45	40	29	26	24	69
	8	4	81	73	54	136	116	101	77	70	62	44	40	37	124
	9	3	110	101	77	180	162	139	102	91	82	59	53	49	169
	10	2	142	133	109	226	206	185	129	116	104	74	67	60	208
	11	1	160	152	133	251	232	211	144	130	117	84	76	70	234
	12	PM	167	160	140	259	242	220	149	134	120	86	79	72	243
	AM														
50 deg North	6	6	26	22	19	46	35	30	30	26	23	17	16	14	31
	7	5	51	45	35	88	70	61	52	47	42	30	27	25	73
	8	4	78	70	51	130	111	95	75	68	60	43	38	36	117
	9	3	102	93	70	167	149	129	95	86	77	55	49	46	155
	10	2	127	116	93	202	184	165	115	103	93	66	60	55	188
	11	1	142	133	109	226	206	185	129	116	104	74	67	60	209
	12	PM	147	137	114	233	214	193	133	120	108	77	70	64	214
U-Volume[c]	UP		53	51	56	1.0	0.9	0.85	1.0	0.9	0.85	1.0	0.9	0.85	
	Heat Flow Down		0.46	0.43	0.40	0.8	0.75	0.7	0.8	0.75	0.7	0.8	0.75	0.7	

[a] ASHRAE GUIDE Values
[b] Curb reflectance 0.7 for solar radiation.
[c] Assumed, no heat flow through curb.
[d] Approximate ratio of width of opening to height of curb.
[e] Normal transmittance = 0.77

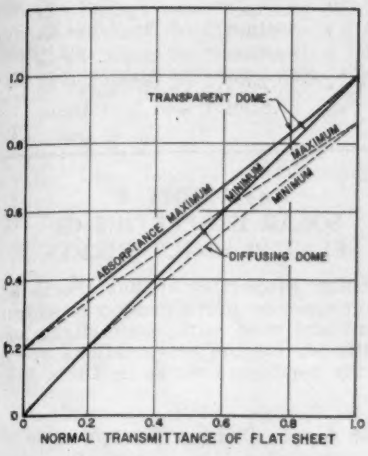

Fig. 8 Effect of solar transmittance, and limiting values of absorptance or reflectance on solar heat gain factors, zero incidence angle

convection-radiation gain. For the clear dome alone, a change in outdoor coefficient from 4 to 1 Btu/hr/sq ft/F will increase the K_{DN} value about 4% at normal incidence. For the low transmittance dome equivalent percentage is 15.

The over-all outdoor coefficient is composed of a convection exchange between the dome and the outdoor air and a radiation exchange between the dome and the sky or adjacent buildings.[3] With the dome at outdoor air temperature there would be no convection exchange with the outdoor air but it could lose heat to the sky by radiation, especially under clear sky conditions.[3] Thus a fixed combined outdoor coefficient only approximates the actual outdoor heat exchanges. In Fig. 13 of reference 1, the observed convection-radiation gain is given for flat glass skylights. This figure shows that, with no solar radiation falling on the flat-glass skylight, it would lose heat at the rate of 8 Btu/hr/sq ft for zero indoor-outdoor temperature difference. Therefore, assuming this figure to be correct, the maximum error in heat gain would be less than 8 Btu/hr/sq ft for the flat glass skylight given in Table II which is based on an outside coefficient of 4. Outside coefficients for flat roofs were reported[11] to vary from 1.14 to 3.82 Btu/hr/sq ft/F and an average value of 3.0 was used.

The U-values which are given in the paper are estimated values which check against observed data which were adjusted for an over-all outside coefficient of 4.0. Estimates were made assuming no heat flow through the curbing which, in effect, was quite low for the well insulated curbing used.

Solar heat gains for skylights other than those tested probably can be estimated from the theory. For a given dome transmittance, it is conceivable that all solar radiation not transmitted is either entirely reflected or entirely absorbed. The transmittance of a skylight is shown by calculation for a given flat sheet transmittance to be slightly higher for zero absorptance (maximum reflectance) than for zero reflectance (maximum absorption). The convection-radiation gain results from solar energy being absorbed by the dome and curbing, and reradiated or convected to the space. For a single dome skylight without curb this amounts to 10 to 20% of the energy absorbed by the dome. The gain due to absorbed solar radiation never exceeded 12% of the solar radiation intensity for any of the skylights tested.

The probably maximum and minimum K-factors for clear and diffusing domes without curb or light diffuser are shown in Fig. 8. For a given transmittance the total solar heat gain is highest for a material having no solar reflectance, and lowest for a material having no solar absorptance. The skylights with curbing and with the diffuser follow the same trends percentage-wise. For diffusing type domes estimates can be made directly from Fig. 7. It should be pointed out that all of the data in this paper apply to a skylight having a square base. For this specific geometry the transmitted component of heat gain will be the same per unit area regardless of size. The convection-radiation component will vary slightly with size but for practical purposes this change can be neglected.

CONCLUSION

The solar heat gains through a number of plastics dome skylights, which are typical of the types in use, can be determined for almost any location and time of the year by use of the design factors which are given in the paper and knowledge of the solar altitude and intensity. Solar heat gains are tabulated for three latitudes for the commonly accepted design day, August 1. Total heat gains for the skylight are determined by combining the solar heat gains with the $U \Delta T$ gains.

The solar heat gain factors are intended to serve as a guide for air-conditioning engineers to better evaluate the instantaneous loads.

ACKNOWLEDGMENT

The authors especially wish to acknowledge the valuable assistance of Dr. O. L. Pierson in supplying needed information and valuable guidance through personal communications. Many Fenn College cooperative students are to be commended for their assistance in detail computations.

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NOMENCLATURE

A Solar absorptance of an element when in combination with other elements of skylight, dimensionless

C_1 Projection of the area of the base of dome for a given incidence angle (See Fig. B-1), area

C_2 Projected area of dome excluding the C_1 area

C_3 Area of dome in defilade from direct sunlight

C + R Convection and radiation gain by room Btu/hr/sq ft

F Shape factor, Fa, b means the portion of energy leaving a which is intercepted by b, dimensionless

$F_{S.\,CRB}$ Portion of direct sunlight passing through the dome which falls on the curb, dimensionless

h Thermal conductance, Btu/hr/sq ft/F

H Height of curb

I Intensity of solar radiation, Btu/hr/sq ft

K Solar heat transfer factor, dimensionless

$L_{\frac{b}{a}}$ Transfer function giving the total average emittance of surface b resulting from initial (before interreflection) average emittance of surface a, dimensionless

R Reflectance for solar radiation including interreflections for itself, dimensionless

R' Reflectance for solar radiation excluding interreflection

T Transmittance for solar radiation including interreflections from itself, dimensionless

T' Transmittance for solar radiation excluding interreflections, dimensionless

t Temperature (F)

S Cross sectional area of curbing or opening area of dome or diffuser, (sq ft)

U Over-all coefficient of heat transfer Btu/hr/sq ft/F

y Convection resistance, (F) (hr) (sq ft) per Btu

z Radiation resistance, (F) (hr) (sq ft) per Btu

Z Thermal resistance, (F) (hr) (sq ft) per Btu

θ Angle of incidence, degrees

ϕ Solar altitude, degrees

α Solar absorptance, dimensionless

Subscripts

- DFR Light diffuser
- DM Dome skylight
- CRB Curbing
- c Convection
- D Direct solar radiation
- d Diffuse or sky radiation
- i Inside
- N Normal to sun's rays
- o Outside
- r Radiation
- s Sun
- 1 Section 1 of Dome see C_1
- 2 Section 2 of Dome see C_2
- 3 Section 3 of Dome see C_3
- 1-2 Section 1 and 2 of Dome

APPENDIX A
SOLAR PROPERTIES OF FLAT PLASTICS SHEETS

Solar Properties of flat sheets as determined by pyrheliometer measurement and used with some slight adjustment, for the computations given in the paper, are shown in Table A-1.

Table A-1 Solar Properties of Flat Acrylic Plastics Sheets

Sheet	(Normal Intensity of Direct Solar Radiation) Incidence Angle, θ	0	30	60	75 Estimated
Clear 3/16 in.	Transmittance	0.86	0.74	0.39	0.14
	Reflectance	0.04	0.04	0.05	0.06
Med Trans 3/16 in.	Trans	0.52	0.45	0.23	0.08
	Refl.	0.36	0.31	0.21	0.14
Low Trans 3/16 in.	Trans	0.27	0.23	0.12	0.04
	Refl.	0.57	0.49	0.31	0.19

Values were selected from curves plotted from experiment points. For the clear plastics the transmittances were computed for incidence angles of 60 and 75 deg.

APPENDIX B
TRANSMITTANCE OF DOMES

Clear Dome The computed transmittance of the clear dome was based on the sunlight passing directly into the room through section 1 of Fig. B1 and the energy entering through section 2 which was diffusely reflected from 3. The reflections from 3 and in turn from sections 1 and 2 of the dome were treated as infinite in number. The transmittance based on incident solar radiation is:

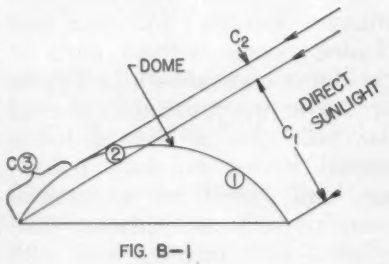


FIG. B-1

$$T_{(DM)} = \frac{\text{(Directly Transmitted Radiation) \& (Radiation Reflected to the Inside)}}{\text{Incident Radiation}} \qquad \text{(B-1)}$$

$$T_{(DM)} = \frac{C_1}{C_1 + C_2} T'_1 + \frac{C_2}{C_1 + C_2} T'_2 R_3 \left(\frac{F_{3,0} + R_{2\text{-}1} F_{2+1,0} F_{3,2+1}}{1 - R_3 F_{3,2+1} R_{2\text{-}1} F_{2+1,3}} \right) \qquad \text{(B-2)}$$

Each of these factors varies with the angle of incidence of the sun. The transmittances and reflectances used in Eq. B-2 are the area-weighted averages, which were obtained by summing the products of zonal areas and the average transmittances or reflectances of the zone over the section considered, and dividing it by the area of the section. The surface of the hemisphere and the segment were divided into zonal areas corresponding to increments of incidence angles of 15 deg. The transmittances and reflectances corresponding to the average incidence angle of each zone were selected from the data of the flat plastics sheets given in Appendix A. These area-weighted solar properties and the transmittance of the dome as given by Eq. B-2 were calculated for solar altitudes from zero to 90 deg.

Since area $C_1 + C_2$ is not readily known, the transmittance for solar radiation measured normal to the sun per unit area of the opening $(C_1 + C_2)_{base}$ is more easily used.

$$T_{(DM)N} = T_{(DM)} \frac{(C_1 + C_2)\,\theta}{(C_1 + C_2)_{base}} \qquad \text{(B-3)}$$

Diffuse Transmitting Domes — The dome was assumed to transmit perfectly diffusely and all reflections also were taken as diffuse.

$$T_{(DM)} = T_{1+2} F_{1+2,0} + T_{1+2} F_{1+2,3} \left[\frac{R_3 (F_{3,0} + F_{3,1+2} R_{1+2} F_{1+2,0})}{1 - F_{1+2,3} R_3 F_{3,1+2} R_{1+2}} \right] \qquad \text{(B-4)}$$

where the first term represents the radiation directly transmitted into the opening and the second term the reflected component.

Clear Dome & Curbing—The direct sunlight passing through the clear dome will fall on the curbing or enter the room. The response-excitation transfer functions were used for these calculations and, for this method of approach, the portion of the solar radiation initially falling on the curbing and reflected from the curbing was assumed as a source of energy which is diffuse in nature. The equation expressing this relationship for direct normal solar radiation is:

$$T_{(DM,\ CRB)N} = T_{(DM)N} (1 - F_{S,CRB}) + T_{(DM)N} F_{S,CRB} R_{CRB} \frac{L_{\frac{room}{CRB}} \alpha_{room} S_{DM}}{R_{room} S_{CRB}} \qquad \text{(B-5)}$$

The first term in the equation is the energy directly transmitted through the dome into the room. The second term is the energy reflected from the curbing which eventually is absorbed by the room after multiple reflections. The transfer functions are based on perfectly diffuse reflections and a plane uniform source of energy. These assumptions only approximate the experimental conditions. The transfer function $L_{\frac{room}{CRB}}$ is the portion of solar radiation originally reflected from the curbing which is rereflected from the room to the skylight. In order that the value of the transfer function be finite, a room reflectance of 0.03 was used.

Diffuse Dome and Curbing—This dome is assumed to transmit only diffusely and thus the transmitted energy is related to the opening or room response from dome excitation.

$$T_{(DM,CRB)N} = T_{(DM)N} \underset{DM}{L_{room}} \times \frac{\alpha_{room}}{R_{room}} \quad \text{(B-6)}$$

Clear-Dome, Curbing, and Light Diffuser—This treatment was similar to that of the clear dome and curbing with additional terms for the diffuser.

$$T_{(DM,CRB,DFR)N} = T_{(DM)N}(1 - F_{S,CRB}) \frac{T_{(DFR)N}}{\cos\theta} + T_{(DM)N} \frac{T_{(DFR)d}}{R_{(DFR)d}}$$

$$\left[\underset{CRB}{L_{DFR}} F_{S,CRB} R_{CRB} \frac{S_{DM}}{S_{CRB}} + \underset{DFR}{(L_{DFR} - 1)} (1 - F_{S,CRB}) R_{(DFR)d} \right] \quad \text{(B-7)}$$

APPENDIX C
REFLECTANCE OF DOMES

For clear domes the reflectance of the clear dome was in part a reflectance from portions 1 and 2, Fig. B-1, and a transmittance through area 3 to the outside of radiation coming in through 2, plus diffuse reflections which pass out through the dome.

The reflectance was taken as

$$R_{(DM)} = R'_{1+2} + \frac{C_1}{C_1 + C_2} T'_2$$

$$\left[T'_3 + \frac{R_3(1 - F_{3,0})(R_{2+1} F_{2+1,3} T'_3 + T'_{2+1}}{1 - R_3(1 - F_{3,0}) R_{2+1} F_{2+1,3}} \right] \quad \text{(C-1)}$$

and for radiation measured normal to the sun's rays

$$R_{(DM)} = R_{(DM)} \frac{(C_1 + C_2)\theta}{(C_1 + C_2)_{base}} \quad \text{(C-2)}$$

For diffuse domes the reflectance both inside and outside of the dome as well as the transmittance was assumed perfectly diffuse.

$$R_{(DM)} = R'_{1+2} + \frac{T_{1+2} F_{1+2,3} (R_3 F_{3,2+1} T_{1+2} + T_3)}{1 - F_{1+2,3} R_3 F_{2,2+1} R_{1+2}} \quad \text{(C-3)}$$

APPENDIX D
ABSORPTANCE OF DOMES

For dome only the absorptance for solar radiation was obtained by subtracting the computed transmittance and reflectance from unity.

$$\alpha_{(DM)} = 1 - T_{(DM)} - R_{(DM)} \quad \text{(D-1)}$$

Clear dome & curbing The response due to excitation from the curbing (direct beam reflected from the curbing) was divided by its own reflectance which results in the total solar radiation falling on that surface. The effective absorptance is merely the product of all energy falling on the surface and the absorptance of that surface. The following absorptances are the percentages actually absorbed of the direct normal incident solar radiation.

Diffuse dome & curbing. With the diffuse dome the only source of energy for the response excitation concept is the dome itself.

Dome absorptance

$$A_{(DM)N} = \alpha_{(DM)N} + T_{(DM)N} F_{S,CRB} R_{CRB} \frac{S_{(DM)}}{S_{CRB}} \underset{CRB}{L_{DM}} \frac{\alpha_{(DM)d}}{R_{(DM)d}} \quad \text{(D-2)}$$

Curb absorptance

$$A_{(CRB)N} = T_{(DM)N} F_{S,CRB} \left[\alpha_{(CRB)} + \underset{CRB}{(L_{CRB} - 1)} R_{CRB} \frac{S_{DM}}{S_{CRB}} \frac{\alpha_{CRB}}{R_{CRB}} \right] \quad \text{(D-3)}$$

Dome absorptance

$$A_{(DM)N} = \alpha_{(DM)N} + \underset{DM}{T_{(DM)N}(L_{DM} - 1)} \frac{\alpha_{(DM)d}}{R_{(DM)d}} \quad \text{(D-4)}$$

Curb absorptance

$$A_{(CRB)N} = T_{(DM)N} \underset{DM}{L_{CRB}} \frac{\alpha_{(CRB)d}}{R_{(CRB)d}} \quad \text{(D-5)}$$

Clear Dome, Curb, and Light Diffuser. Two possible sources of energy exist with this combination. These are the reflections of the direct beam from the curb and from the light diffuser.

Dome absorption

$$A_{(DM)N} = \alpha_{(DM)N} + \frac{\alpha_{(DM)d}}{R_{(DM)d}} T_{(DM)N} \left[F_{S,CRB} R_{CRB} \frac{S_{DM}}{S_{CRB}} \underset{CRB}{L_{(DM)}} + (1 - F_{S,CRB}) R_{(DFR)d} \underset{DFR}{L_{DM}} \right] \quad \text{(D-6)}$$

Curb Absorption

$$A_{(CRB)N} = \alpha_{(CRB)N} T_{(DM)N} F_{S,CRB} + \frac{\alpha_{(CRB)d}}{R_{(CRB)d}} T_{(DM)N} \left[F_{S,CRB} R_{CRB} \frac{S_{DM}}{S_{CRB}} \underset{CRB}{(L_{CRB} - 1)} + (1 - F_{S,CRB}) \times R_{(DFR)d} \underset{DFR}{L_{CRB}} \right] \quad \text{(D-7)}$$

Diffuser Absorption

$$A_{(DFR)N} = \alpha_{(DFR)N} T_{(DM)N}(1 - F_{S,CRB}) + \frac{\alpha_{(DFR)d}}{R_{(DFR)d}} \left[(1 - F_{S,CRB}) T_{(DM)N} R_{(DFR)d} \underset{DFR}{(L_{DFR} - 1)} + F_{S,CRB} R_{CRB} \underset{SCRB}{\frac{SDM}{}} \underset{CRB}{L_{DFR}} \right] \quad \text{(D-8)}$$

APPENDIX E
CONVECTION-RADIATION GAIN

Single dome—Here the convection and radiation gains by the space were analyzed according to Fig. E-1 and the

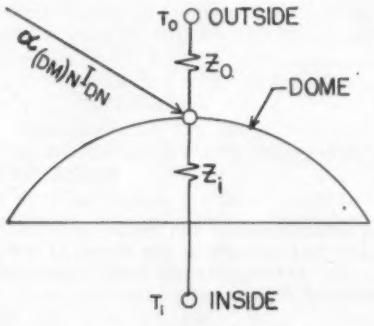


FIG. E-1

results compared with the experimental data. The solar energy absorbed by the dome $\alpha_{(DM)N} I_{DN}$ was obtained from the conversion of equation D-1 to give the absorption for direct normal radiation. For $t_o = t_i$, the convection and radiation gain to the inside is

$$C\&R = \frac{Zo}{Zo + Zi} \alpha_{(DM)N} I_{DN} \qquad (E\text{-}1)$$

where the reciprocal of the thermal resistance is conduction

$$C\&R = U\left(\frac{\alpha_{(DM)N} I_{DN}}{h_o}\right) \qquad (E\text{-}2)$$

including the temperature effect

$$C\&R = U\left(\frac{\alpha_{(DM)N} I_{DN}}{h_o} + to - ti\right) \qquad (E\text{-}3)$$

The U-value will vary with temperature difference and mean temperature and consequently is somewhat dependent upon the absorbed solar energy. For general application the variation is neglected.

Dome and curbing. The thermal interchange resistance for the case of a dome and curb are shown in Fig. E-2. Three resistances marked z represent thermal radiation exchanges between the surfaces, and the three resistances y represent the convection exchange between surface and air. The curbing was assumed to be adiabatic for this analysis. The resistances marked z were first converted to a delta circuit, then combined with the delta circuit of the radiation resistances and the combination changed to the wye circuit shown in Fig. E-3. From this relation the connection and radiation gain to the inside is:

$$C\&R = \frac{1}{Zo + Zc + Zo}$$
$$[Zo\, A_{(DM)N}\, I_{DN} +$$
$$(Zo + Zc)\, A_{(CRB)N}\, I_{DN} + to - ti] \qquad (E\text{-}4)$$

$$C\&R = U\left[\frac{A_{(DM)N}\, I_{DN}}{h_o} + \right.$$
$$\left.(Zo + Zc)\, A_{(CRB)N}\, I_{DN} + t_o - t_i\right] \qquad (E\text{-}5)$$

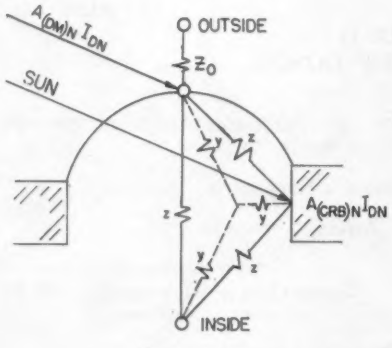


FIG. E-2

Dome, curbing, and light diffuser. The thermal circuit for this combination is similar to that shown in Fig. E-2 with the junction labeled "inside" becoming instead the diffuser. Also a thermal resistance must be added connecting the diffuser with the inside.

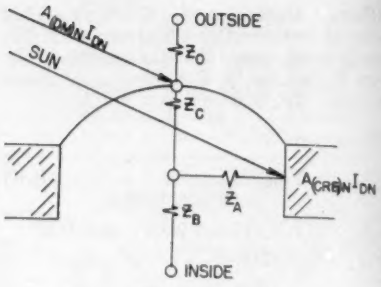


FIG. E-3

$$C\&R = \frac{1}{Zo + Zc + Z_B + Zi} [$$
$$[Zo\, A_{(DM)N}\, I_{DN} +$$
$$(Zo + Zc)\, A_{(CRB)N}\, I_{DN} +$$
$$(Zo + Zc + Z_B)$$
$$A_{(DFB)N}\, I_{DN} + t_o - t_i] \qquad (E\text{-}6)$$

The range of convection and radiation coefficients shown in Table E-1 were selected for evaluation of the thermal circuits.

The convection coefficients for the dome and light diffuser were obtained from experiments and are values corresponding to temperature differentials of 10 and 30 F. The convection coefficient for the curbing was determined from reference 10 of the paper. For the calorimeter the convection was determined from computation and experimental data from flat glass tests.

The inside resistance was taken as unity for the heat flow down. Where the diffuser was utilized, the convection coefficients for heat flow upwards were used; but for the dome and curb, the heat flow down values were used. In all cases the outside resistance Zo was taken as 0.25 which corresponds with an overall heat transfer coefficient of 4 Btu/hr/sq ft/F.

Table E-1 Radiation & Convection Coefficients

	Radiation Coef. hr	Convection Coefficient h_c	
		UP	DOWN
Dome	0.7-1.0	0.72-0.90	0.15-0.17
Curb 9' x 18'	0.7-1.0	0.60-0.85	0.60-0.85
Light Diffuser	0.7-1.0	0.61-0.76	0.13-0.14
Calorimeter or Room	0.7-1.0	0.45-0.48	0.17-0.18

ASHRAE
NATIONAL MEETINGS
AHEAD

1962
Jan. 29-Feb. 1 Semiannual
St. Louis, Mo.

June 25-27 69th Annual
Miami Beach, Fla.

1963
Feb. 11-14 Semiannual
New York, N. Y.

June 24-26 70th Annual
Milwaukee, Wisc.

Board Stand on Laboratory Reaffirmed at Denver Meeting

Putting an end to the agitated question of whether or not there should be reconsideration of the earlier Board of Directors' decision to close the ASHRAE Research Laboratory in Cleveland,* overwhelming support by members of a clarifying resolution introduced at the 68th Annual Meeting in Denver by Research and Technical Committee Chairman E. P. Palmatier has reduced the matter to routine close-out operations. These approach closely the tentative schedule established originally.

At the opening session on Monday Chairman Palmatier proposed:

WHEREAS the Board of Directors, in accordance with its duty under the By-laws to manage the affairs of the Society, has taken action to close the Laboratory in Cleveland, and

WHEREAS subsequent objection to this action by some members has created confusion which is handicapping the effective operation of the Research and Technical Committee in advancing the Research Program, and

WHEREAS we have confidence that the Board of Directors has acted, and will continue to act, in the best interests of the Society and a vigorous research program,

WHEREAS the Research and Technical Committee has diligently carried out the decision of the Board of Directors regarding closing of the Laboratory, in spite of the fact that on Jan. 5, 1961, the R & T Committee went on record in disagreement with the decision of the Board;

WHEREAS subsequent planning and developments have led the R & T Committee to believe that the only logical course the Society can take is to continue in our plans to close the Laboratory and plan our research program with cooperative institutions;

NOW THEREFORE the Research and Technical Committee wishes to go on record as supporting the decisions of the Board of Directors with regard to the research program.

Then, after a number of members had offered their viewpoints from the floor, the measure was put to a member vote. The motion was carried 4694 for, 31 against; three invalid votes.

The officer slate for 1961-62 was formally voted in as were all of the proposed amendments to the By-laws.

Denver ranks high among the best of our well-handled and rewarding Society meetings. The technical program was varied and participated in widely by those attending. The heavy social program included the Welcome Luncheon, the Annual Banquet, the Western Hospitality Evening and the Flagstaff Mountain Cook-Out Party to balance the significant developments covered at Technical, Symposium and Forum sessions.

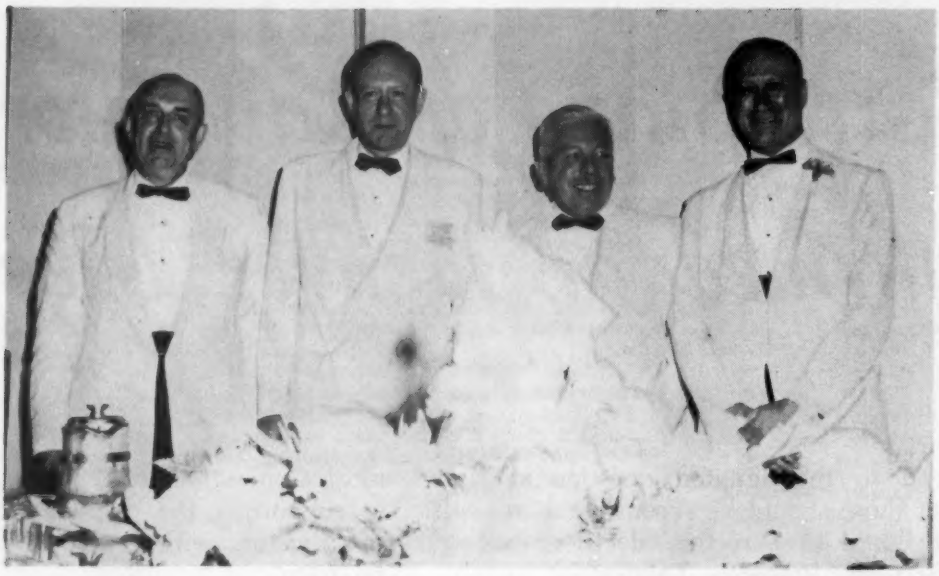
At the Denver Meeting, Warren S. Harris, C. O. Mackey, George V. Parmelee, Alfred J. Offner, G. L. Tuve, Sewell H. Downs and John E. Haines were made Fellows.

The F. Paul Anderson Award, highest of ASHRAE honors, this year went to J. Donald Kroeker. Further details of awards and honors appear elsewhere in this issue.

There were 648 registered members and male guests this year, as compared with 384 when predecessor ASHAE met in Denver in 1953. Total registration exceeded 800.

At the First Technical Session – Steam vacuum refrigeration, Elliot Spencer related, was first investigated prior to 1901 and applied limitedly at that time; however, restrictions of early ejector design and antiquated pumps and controls prevented its widespread use. Despite these shortcomings, steam vacuum refrigeration systems were applied successfully and, according to Author Spencer, some systems are still in operation following 20 or 30 years of service.

* See the ASHRAE JOURNAL, January 1961 (pages 54-56), April 1961 (pages 70-71), May 1961 (pages 74-77) and June 1961 (pages 66-68).



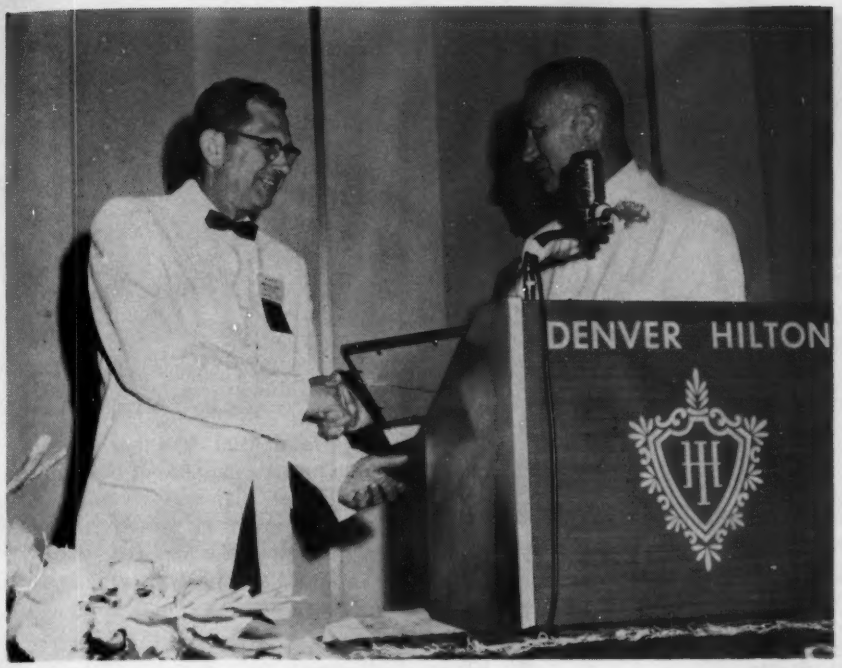
The new team. Treasurer John E. Dube. Second Vice President Frank H. Faust. First Vice President John H. Fox. President John Everetts, Jr.

Under the Chicago-inaugurated plan for the induction of new officers, Presidential Member John E. Haines so served at Denver. Here is the moment when John Everetts, Jr., became President of ASHRAE

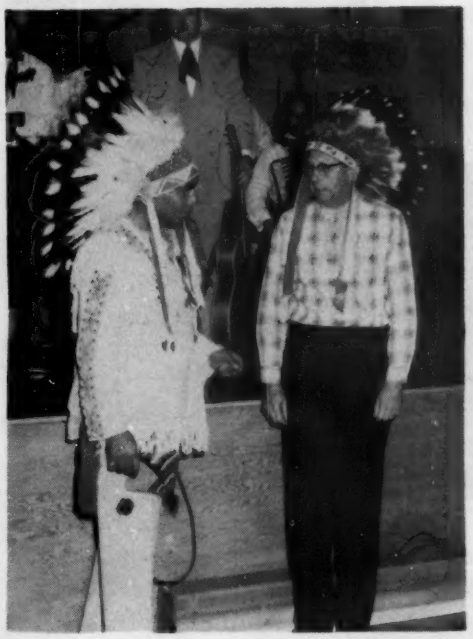
Secretary Sidney Sellers of the Rocky Mountain Host Chapter was most active in his efforts to assure a successful and enjoyable meeting. Here shown at the Banquet

To J. Donald Kroeker went the F. Paul Anderson Award, ASHRAE's highest honor, consisting tangibly of a certificate and a gold medal; here being presented by outgoing President Tull

Preceding the induction of the new slate of officers, President Tull formally addressed the Banquet

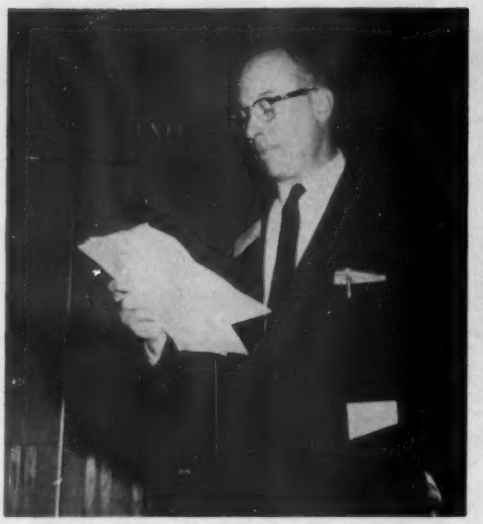


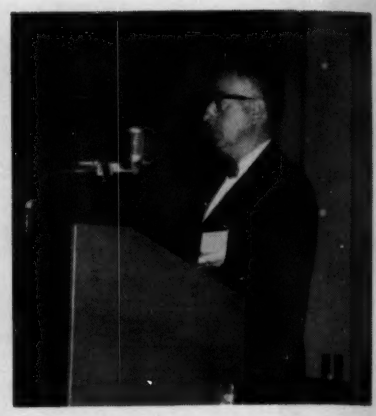
Newly-installed President John Everetts, Jr., presented a Certificate of Appreciation and a gift to outgoing President R. H. Tull. Presentation of a remembrance was made to Mrs. Tull



President R. H. Tull was made an Indian chief, having been duly presented with the appropriate headdress at the Western Party on Monday night. Indians, in full costume, representing 19 tribes, participated in formalized ritualistic dances

At the General Assembly and Business Session, which opened the 68th Annual Meeting, Chairman E. P. Palmatier of the Research and Technical Committee introduced a resolution (presented in full elsewhere in this news report) backing fully the earlier action taken by ASHRAE to close the Cleveland Research Laboratory. Considerable floor discussion followed before the measure was put to a vote. The action of the Board of Directors received overwhelming approval at this time

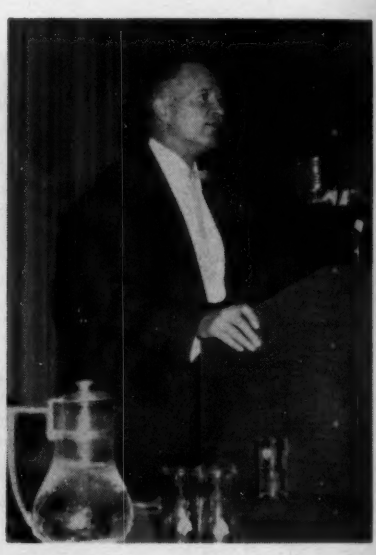




Presidential Member Elmer Queer, a member of the Long Range Planning Committee, the report of which recommended the closing, reiterated the fundamental reasons for doing so

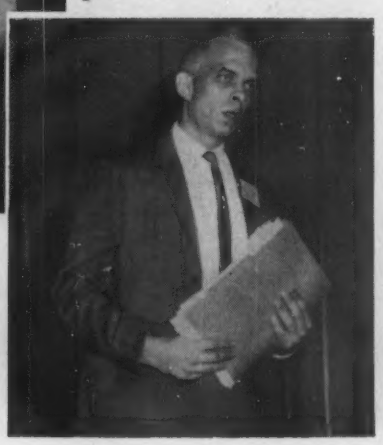
Vice Chairman Neil B. Hutcheon reassured as to the to-be-anticipated future for ASHRAE non-Laboratory research

Fellow Ralph M. Westcott provided another aye



Counsel S. B. Vreeland reviewed the specific legal aspects

John H. Clarke and H. T. Gilkey offered the focus of the opposition



Re-voting of some of the originally mailed proxies by those present and a floor vote were parts of this spirited session

It was stated that improvements in accessory design and ejector efficiency have taken place during the past 25 years which have resulted in the application of these systems in increasing numbers for industrial and commercial water chilling requirements.

Nevertheless, the conventional steam vacuum refrigeration cycle has not been applied extensively, despite its many advantages, including reliability (no moving parts), low maintenance, use of water only as a refrigerant and the ability to use either high or low pressure steam. The reason for this, as explained, relates to the fact that as the available condenser water temperature rises, both the steam rate and the condenser rate increase. This means that unless large quantities of condenser water are available at low cost, as from a river or other large body of water, or unless water is available at a low temperature, as from a well, cooling water requirements will substantially exceed that of the other refrigeration systems. When water from a cooling tower is used at the conventional 85 F level, steam vacuum refrigeration will have an indicated full load steam rate and water rate somewhat higher than either centrifugal refrigeration or lithium bromide absorption, it was pointed out.

Thus, the author asserted that any modification of the steam vacuum refrigeration cycle which reduces its steam consumption and eliminates the need for condensing water in large quantities will affect to a large extent the feasibility and usefulness of this system for comfort air conditioning applications. Mr. Spencer indicated that by replacing the water-cooled condenser of conventional steam vacuum refrigeration with a wetted air-cooled surface condenser, steam consumption will be reduced and water requirements lowered to fractional levels. Additionally, the need for large condenser water recirculating pumps is eliminated.

As to size, the modified steam vacuum system described occupies approximately the same space normally required for the cooling tower alone for conventional steam-operated systems. Dimensions for a typical 225-ton, 45 F modified steam vacuum system were designated as approximating those for a 300-ton, 50 F chilled water system. Space requirements for larger tonnages would not necessarily increase proportionately, the author indicated.

It was brought out that while weight requirements of the modified steam vacuum system are greater than the weight of the cooling tower alone for steam-operated systems, the weight of the entire system is about equal to or slightly less than the combined weight of a cooling tower and steam-operated refrigeration system where the entire system is installed on the roof. If the modified steam vacuum system must be located on a lower floor set-back, the system loses some of the available chilled water pumping head because the returning chilled water is flashed to the vacuum in the evaporator. Closed chilled water cycles available in other systems do not have this disadvantage.

Summarizing, Author Spencer declared that this modified steam vacuum refrigeration system is suitable for comfort air conditioning — a system with low installed first cost, low annual steam consumption, one which is reliable and maintenance free, although limited in size at the present time to systems under 500 ton of refrigeration in a single package.

In their presentation, Authors Thomas M. Olcott and Harold A. Blum explained that the principle of substituting a refrigerant expansion engine for the expansion valve of a vapor compression cycle system is quite well known and correspondingly documented. However, performance data for such an engine have been lacking.

As part of their investigation, the authors constructed a simple engine, tested it and compared its performance with the isentropic expansion. It was reported that with this device the recovery of available energy was only as high as 10%; but the analysis did reveal that the pressure drop and leakage in the valves were major causes of the low energy recovery. Were these eliminated, it was stated that the percentage recovery could have reached 40%. The authors also made a theoretical comparison between the single circuit refrigeration system, utilizing the expansion engine and the cascade system.

Results of this investigation pointed out that the engine described demonstrated successfully the principle of the refrigerant engine; although it was conceded that the fraction of isentropic process available energy was too small to be commercially interesting.

The authors' conclusion was that, in order to make this system a definite contribution to the art, this principle should be applied to an operating refrigeration cycle in such a way that work obtained from the engine be used either to reduce the work input to the compressor or handle auxiliary equipment.

Strength characteristics of ice at the interface was the subject of J. K. Stene and W. E. Fontaine, who reported that in previous research to determine the adhesion of ice formed on various surfaces, the strength has been calculated based on the average stress required to separate the ice and the surface in either tension or shear. In their present investigations, Authors Stene and Fontaine found that when ice was formed between two cold metal surfaces, the ice

failed at a crystal interface. The cause for this was attributed to ice formation advancing on fronts from both surfaces. It was found that if one surface was left warm, then this type of failure was eliminated. However, the breaking plane in the tensile tests always occurred in the ice, but the breaks moved closer to the surface that was originally cold.

As reported, test results in tension were put together for all clean metal surfaces. Metals used were brass, copper, aluminum and steel. Average tensile strength was given at about 105 psi for ice greater than ⅛ in. thick. If tests were performed varying the thickness of the ice, it was found that the stress increased as the thickness decreased.

In another series of tests, the authors continued, some metal surfaces were treated with silicone compounds. It was cited that the average value of the tensile stress for all tests was 8 psi. A tendency for increasing stress as the number of removals from one surface was extended was attributed to the removal of the silicone surface with the ice. The authors related that in this series complete removal of ice was achieved. In the shear tests, the shearing strength of adhesion was determined to be less than the tensile strength of adhesion, averaging about 65 psi and all separations occurred at the interface and not in the ice.

Authors Stene and Fontaine made two generalizations based upon their findings – namely, that the area of real contact is important in determining adhesion forces and that there are no smooth surfaces, since all carefully polished surfaces have peaks and depressions which are large compared with molecular dimensions. According to the authors, the roughness of a surface can cause strong and weak points in the interface due to the adhesion forces of near points.

As a result of the investigations, it was determined that ice adheres most strongly to the anodized aluminum surface, next to stainless steel and then weakly to Teflon. It was disclosed that ice will adhere more strongly to an oxidized metal surface; since the anodized surface was a completely oxidized surface, this statement was verified. The stress for metallic surfaces is greater than for organic surfaces and the stress for the completely oxidized surface is greater than that for the incompletely oxidized surface, it was pointed out.

In conclusion, the authors stated that work carried out in this experiment, including the preliminary phases, confirmed that ice is not a uniform structure of crystals with an unusual strength of adhesion; rather, that ice is formed with imperfections between crystals that make weak spots, which influence the results obtained from strength tests. On the average, there is a strength of adhesion which is a characteristic of the material to which the ice is frozen and which is a function of the temperature. The authors expressed the opinion that tests carried out in the same manner will provide results within the same limits determined in this specific experiment. Again, it is held that the equations derived may be extrapolated beyond the range involved. According to observations gathered from this experiment, if carried above 10 F, slow sliding will be present before the sudden failure. Finally, it was indicated that the prediction for the failure strength of thicker samples of ice is also questionable, since the larger volumes provide greater possibility of flaws in the ice which will produce different results.

With heat and mass transfer in dehumidifying surface coils as his topic, W. L. Bryan opened his presentation by explaining that in an air conditioning heat exchanger, such as a cooling coil, both heat and water are removed from the air. The added complication of vapor diffusion together with the transfer of heat means that a proper method for rating and making performance calculations for the heat and mass transferred must be found by the air conditioning and refrigerating engineer. Extensive research and analysis has been accorded this problem, yet the fundamental basis for the methods now in use is not altogether acceptable, according to the author, who explained that in an attempt to achieve simplicity in application, assumptions have been made or empirical equations proposed. In addition to this loss in accuracy, most notably if the operation conditions differ from the conventional range, the basis for the assumptions has not been fully justified, it was pointed out.

Experimental data for the heat and mass transfer on a bare tube surface coil were presented. Experimentally determined coil surface temperature provided a direct evaluation of the heat and mass transfer coefficients. As reported, this measured surface temperature was found to be much lower than would be expected from the present methods for surface temperature calculations.

It was indicated that the coefficient of heat transfer for dry cooling was lower than when the coil was operated with dehumidification. The author stated that it would seem likely that the friction factor would be lower also for dry cooling; however, the core pressure drop taken with these data was quite small and the accuracy of the measurement was not sufficient for conclusions.

At the Second Technical Session – Defining a

flexible orifice as one in which the orifice area varies with pressure, R. D. Ulrich, W. E. Fontaine and O. W. Witzell explained that these orifices are of interest because of the large variety of flow rate-pressure drop relations which may be obtained by various designs of the orifice and the method of supporting the orifice. The phenomenon of decreasing flow rate with increasing pressure drop suggests the use of these orifices as devices for the regulation of flow, the authors pointed out. Also, there have been efforts to use these orifices in refrigeration systems to yield maximum flow rates at design conditions and to decrease the flow rate with increasing pressure differences. The authors observed that in earlier investigations power consumption in a refrigeration system was compared with expansion occurring through capillary tubes at high condenser pressures.

The work described in this paper was undertaken with the objective of determining the variables which influence the flow. Following determination of these variables, the problem was then one of correlation. The following conditions prevailed: the geometries were short, right cylindrical orifices with centered, cylindrical holes; orifices were made of rubber; support plates were flat with circular holes at least as large as the orifice; the fluid flowing was water at room temperatures; only steady flow was studied.

The authors reported on a method developed for predicting the flow rate-pressure drop relationship for a given geometrical situation and rubber hardness for a flexible orifice. It was stated that this method could be expanded to include a large variety of similar orifices, and possibly could be extended into various geometries and orifice materials. For thin orifices the flow coefficients were found to be independent of Reynolds Number; for thick orifices the flow coefficient increased as Reynolds Number increased.

The authors declared that in many cases the difference between the predicted and measured flow rates was less than 10% of the measured values; the maximum difference found was less than 25%. Because of the complexity of the problem, the authors are of the opinion that these comparisons are acceptable at this state in the solution of the problem of flexible orifices. The region where the mass flow rate vs. pressure drop relation had a negative slope was unstable. It was pointed out that in each case there was either cavitation of the fluid or vibration of the orifice or both, however, it is not known which of the two phenomena (vibration, cavitation) was the cause or effect.

Claus J. Bushouse spoke of a new insulation system, for use in stators applied to hermetically sealed refrigeration and air conditioning systems, which consists of acrylic coated magnet wire, polyester film insulation and polyester covered lead cable and polyester lacing cord. According to the author, this system has highly desirable qualities, such as higher temperature capabilities, non-water forming, and the possibility of shorter, higher temperature dehydration.

However, it was pointed out that this system has been found to be vulnerable to methanol. The point of alcohol attack is the polyester film sheet insulation, polyethylene terphthalate. At elevated temperatures methanol was found to degrade the film to the point of embrittlement. This was established by sealed tube work and compressor testing.

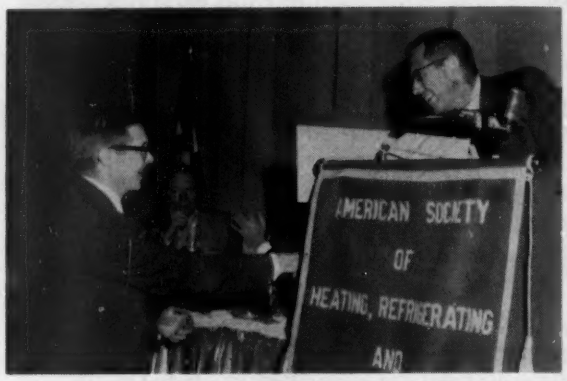
Author Bushouse noted that methanol is used as a remedial measure in hermetic refrigeration units; therefore, various compounds were studied in order to determine their reactivity with polyethylene terphthalate film used in the new insulation system. As reported, preliminary work showed that methanol is quite reactive, causing rapid embrittlement of the film. Studies of each alcohol with the film in a specially designed glass tube indicated that methanol and four other primary alcohols have the same order of reactivity; that two secondary alcohols were about equal to water in reactivity; and that tertiary butyl alcohol had no reactivity. It was explained that no recommendations can be made since the most likely substitute, isopropyl alcohol, is just as reactive as water, it might fall short in antifreeze properties and it might cause an increase in non-condensable gases when used in the hermetic system. Author Bushouse summarized that any of the ordinary antifreeze alcohols when added to the system would cause rapid degradation of the polyester film in this new insulation system.

Since absorption cooling cycles use a combination of absorption and stripping steps instead of the mechanical compressor of the vapor compression cycle, pointed out Allen Thieme and Lyle F. Albright, advantages of the absorption cycle include lower operation costs if efficient absorption and stripping steps can be realized. Efficiency of these steps is determined to a considerable extent by the chemical and physical properties of the solvent-refrigerant combination; therefore, evaluation of these properties is of importance in the development of an economical absorption cycle, it was indicated.

According to the authors, from the standpoint of the absorption cooling cycle, an efficient cycle would be one in which a large amount of refrigerant is evaporated in the generator from a given amount of solution. Such a solvent-

Awards at the Welcome Luncheon went to B. P. Morabito for his paper, "How Higher Cooling Coil Differentials Effect System Economies" (JOURNAL, August 1960); to G. V. Downing, Jr., for his "Thermoelectric Materials" (JOURNAL, November 1960); and to C. F. Speich, who co-authored "Acoustical Systems Determine Oil Burner Pulsations and their Amplitudes" (JOURNAL, November 1960). President R. H. Tull officiated

B. P. Morabito receives the Wolverine-ASHRAE Diamond Key Award



To G. V. Downing, Jr., went the ASHRAE Klixon Award

ASHRAE's Willis H. Carrier Award was made to C. F. Speich

At the Welcome Luncheon featured speaker was President R. T. Person of the Public Service Company of Colorado, who extended a warm welcome to all to Denver

President L. R. Bindner of the Rocky Mountain Host Chapter spoke at the Welcome Luncheon

As General Chairman V. J. Johnson extended his greetings

Chairman R. A. Line of the Program Committee introduced each of the speakers of various sessions at the Luncheon

refrigerant system would be one in which the refrigerant is soluble in the solvent in large excess of the amount predicted by Raoult's Law.

Authors Thieme and Albright reported upon an investigation where the solubilities of three fluorochloromethanes were studied in several organic compounds containing a nitrogen atom. Additionally, a preliminary attempt was made to evaluate binary mixtures of low volatile organic liquids as solvents.

Several nitrogen-containing organic solvents were tested; it was found that some amines, nitriles and formamides were good solvents for Refrigerants 21 and 22, which contain a hydrogen atom and therefore can hydrogen bond. It was reported that the two best solvents were dimethyl formamide and diethyl formamide, which produced mixtures with C.O.P. about 15 and 2% better than mixtures containing the dimethyl ether of tetraethylene glycol. C.O.P. values ranging from 0.27 to 0.44 were estimated for mixtures of the formamides.

Although several binary mixtures were investigated as solvents for the refrigerant, the authors pointed out that none of those tested showed improved solubility characteristics as compared to the pure compounds.

In their presentation, R. C. McHarness and D. D. Chapman noted that in engineering with the relative cooling capacities of various compounds and azeotropes, tables of thermodynamic properties or pressure-enthalpy diagrams can be used to calculate theoretical values for the individual refrigerants and to compare data at any given set of operating conditions. However, this does not take into consideration the behavior of the compressor and other components of an operating system and thus does not give the practical answer desired, it was indicated. Unit operating tests were cited as needed to obtain such values.

The authors observed that much of the work carried out thus far in the field of mixed refrigerants relates to the increased capacity obtainable with mixtures and the heat transfer problems and advantages involved in their use. It was stated that the purpose of the present stutly was to expand the information available on azeotropes and mixtures and to compare their refrigerating capacity and operating data with similar data for pure compounds. The authors carried out most of the tests at nominal evaporator temperatures of −20, −10 and 40 F and a nominal condenser temperature of 65 F. Additionally, a few runs were made at nominal evaporator temperatures as low as −60 F.

Authors McHarness and Chapman described the performance of a variety of refrigerants, azeotropes and mixtures which were studied in a calorimeter unit. Comparative capacity data were obtained over a range of temperatures typical of refrigerating and air-conditioning applications. It was reported that these studies showed that the use of mixtures is both feasible and practical and that definite and reproducible operation can be obtained with them. The authors expressed the opinion that actual operating information provides a basis for further consideration.

The authors were confident that this practical demonstration of the use of mixed refrigerants provides almost unlimited possibilities for varying the capacity of refrigerating systems. It was pointed out that design engineers will have greater flexibility in studying new applications and problems and in selecting compressor size and speed. In closing, the authors remarked that further studies with refrigerant mixtures are continuing, as well as the search for new refrigerants.

At the Third Technical Session — L. F. Schutrum and N. Ozisik, who reported upon a study of solar heat gains through domed skylights, explained that while solar heat gains through four types of light controlling and three types of flat glass skylights were reported upon previously, information on the heat gains through a plastics dome skylight, which is becoming increasingly popular, has not been available. The study which the authors carried out produced data useful to the air conditioning engineer in determining solar heat gains through acrylic plastics dome skylights. Three domes having high, medium and low light transmitting qualities were selected to cover the range of normal practice, and a fourth dome, which was painted black, was tested in the ASHRAE Solar Calorimeter. This calorimeter has been used for some time to compile data on heat flow through fenestration under the guidance of the Society's Technical Advisory Committees, it was indicated. The present study was started under the guidance of the former TAC on Heat Flow through Fenestration.

As a result of their studies, the authors contended that solar heat gains through a number of plastics dome skylights, which are typical of the types in use, can be determined for almost any location and time of the year by use of the design factors presented within this paper and knowledge of the solar altitude and intensity. Solar heat gains were tabulated for three latitudes for the commonly accepted design day, August 1. It was demonstrated that total heat gains for the skylights were determined by combining the solar heat gains with the $U\Delta T$ gains. The authors cited solar heat gain factors as intended to serve as a guide for air-conditioning engineers to better eveluate the instantaneous loads.

Since air infiltration through doors may cause discomfort in areas near the doors, it is desirable to know the rate of infiltration for proper sizing of the heating and cooling equipment, pointed out L. F. Schutrum, N. Ozisik, C. M. Humphreys and J. T. Baker. They disclosed that recent research has supplied comprehensive information on air infiltration through swinging doors, but that only limited data are available on air infiltration rates through revolving doors. An investigation which sought to provide additional information on this subject was undertaken by the authors at the ASHRAE Research Laboratory under the guidance of the former Research Advisory Committee on Heating and Air Conditioning Loads. Experiments were made both under heating and cooling conditions. With the motor-driven revolving door, the door speed and temperature and pressure differences between indoors and outdoors were recognized as significant factors influencing the air infiltration rate. With manually operated doors, additional information regarding a representative traffic pattern and the corresponding average door rpm was needed, it was pointed out. The amount of air turbulence inside and outside and the condition of the door seals would influence infiltration to a certain extent. This research program attempted to determine the influence of all of these variables on air infiltration.

On the basis of their findings, the authors reported that infiltration through a revolving door can be estimated by combining the air leakage past the door seals with the infiltration related to the revolving of the door. The magnitude of air leakage through the seals of the door was cited as the result of the pressure differential across the building entrance and the size of the openings at the seal. It was stated that revolving the door causes an exchange of indoor and outdoor air of approximately equal volume. The amount of this air exchange depends upon the door speed and the temperature differential and somewhat upon the wind and indoor air velocities.

Infiltration due to air leakage past the seals of the door was indicated by the authors as a function of the indoor-outdoor pressure differential. Infiltration related to the door was presented for a motor-driven revolving door and for a manually operated door for traffic rates up to 2000 people per hour. It was pointed out these data are based on the use of door seals which provide good contact with the adjacent surfaces. If seals deteriorate to the point that good contact is not maintained, leakage past the seals will increase greatly, the authors emphasized.

Study of air conditioning odors has been an important part in the environmental program at the ASHRAE Research Laboratory and, in his paper, William F. Kerka described a procedure used to evaluate odor control methods such as odor modification, sorption and oxidation. It was pointed out that control of odors within air conditioned spaces has become an increasingly important factor in enclosed environments, especially where the use of fresh, outdoor make-up air is reduced to a minimum and the recirculated indoor air picks up odors that are generated in the occupied spaces.

The author reported that the procedure of testing described can be applied to a number of odor control methods. It was indicated that the main deficiency of the static tvpe of test, which was used, is that absolute stability of the conditions may not exist during the treatment and evaluation periods. While significant reductions in malodor level can be achieved by treatment with odor modifiers, it was stated that the odor of the modifier also is perceptible.

The following conclusions were reached by the author: that significant reduction in malodor level can result from treatment with activated carbon and that reduction of odor levels by ozone at concentrations slightly above threshold is not statistically conclusive by the results presented in this study.

Author Kerka emphasized that the procedures described should be considered exploratory in nature and should serve as a guide for further experimentation in the broad field of odor control.

Benjamin Y. H. Liu and Richard C. Jordan has as their subject daily insolation on surfaces tilted toward the equator. They observed that measurements of solar radiation have been made up to this time primarily on horizontal surfaces and data for sloped or vertical surfaces are not readily available. However, it was pointed out, the solution of engineering and scientific problems involving solar radiation generally requires a knowledge of the solar radiation incident upon surfaces of various orientation. The authors developed a method which enables the solar radiation on sloped surfaces to be determined when the radiation incident upon a horizontal surface is known.

It was shown that although the method proposed (with modification) may be applicable to surfaces of any orientation, the present research was restricted to surfaces tilted toward the equator only; that is, tilted toward the south for localities in the northern hemisphere and tilted toward the north for localities in the southern hemisphere. It was stated that solar energy collecting devices of current interest, with the exception of those equatorially mounted to follow the diurnal motion of the sun and those at

outer space, are generally orientated for maximum exposure to sunshine. Radiation of solar origin only is considered and thermal radiation, either from the ground or from the atmosphere, is not involved, according to the authors.

Though the subject of the presently reported research had been investigated previously, the resulting empirical treatments of the problem differ from the fundamental approach followed at this time.

Since 1945, the U. S. Weather Bureau has made measurements at its solar radiation station, at Blue Hill, Mass., of the total radiation incident upon a south facing vertical surface and a horizontal surface. These experimental data were used by the authors.

As reported, experimental conversion factors for each month were obtained by first classifying the days into a number of groups so that the values of H (daily summation of total radiation incident upon a horizontal surface) for days belonging to each group fall within a limited interval of values (the sizes of the intervals are indicated by the differences of the values of H and of the neighboring points) and the ratio $R = H_t/H$ (R = conversion factor; H_t = daily summation of total radiation incident upon a tilted surface) is then computed for each group.

It was indicated that during the winter months when the positions of the sun in the sky are more favorable for the incidence of the direct solar radiation on a vertical surface facing south and consequently the values of R_D (conversion factor for the direct reflected radiation) are large, the conversion factor for daily total radiation increases rapidly with decreasing atmospheric cloudiness due to the decrease of the ratio D/H. (D = daily summation of the diffuse radiation incident upon a horizontal surface.) On the contrary, for the month of June, when R_D is small, the value of R decreases slightly when K_T (cloudiness index) increases. It was brought out that this does not mean that during the month of June the radiation incident upon a south facing vertical surface at Blue Hill becomes less as the atmosphere becomes clearer. According to the authors, the radiation incident upon the vertical surface is the product RH and is not determined by the value of R alone.

At the Fourth Technical Session — According to J. Richard Wright and Edward J. Brown, residential summer cooling applications frequently require installation of ductwork in non-air conditioned spaces, such as basements, crawl spaces and attics. The result is that the conditioned air within the ducts is subject to heat gain from the non-conditioned spaces, since the ambient temperature may vary from 75 F in basements to 140 F or more in attic spaces.

In order to conduct an experimental investigation of heat gain to air flowing in ducts, these must be enclosed in a space throughout which a uniform air temperature can be maintained, but considerable expense would be involved to provide such a space in a laboratory. As an alternate means of studying this problem, the ductwork in the basement and attic of the Warm Air Heating Research Residence No. 2 at the University of Illinois was provided with sufficient instrumentation to measure the average temperature rise of the conditioned air. The authors noted that this instrumentation was not considered suitable for determination of heat gain by the differences in enthalpies at two stations in a duct; this requires an evaluation of temperature and velocity profiles to determine total enthalpies at each station. The Residence installation did not provide a means of controlling the ambient air temperature of the basement and attic; however, it was possible to measure the average temperature rise in the ducts after the summer air conditioner had been in operation for several hours and steady-state conditions prevailed within the basement or attic.

On the basis of this study, Authors Wright and Brown offered an analysis of the effects of duct air temperature and velocity and ambient air temperature on the temperature rise of air flowing in round ducts used for summer air conditioning. This analysis was based on the following assumptions: the heat transfer rate is constant, does not vary with time; the mean radiant temperature is equal to the ambient air temperature; and condensation does not occur on the exterior surface of uninsulated ducts or on the exterior vapor barrier surface of insulated ducts.

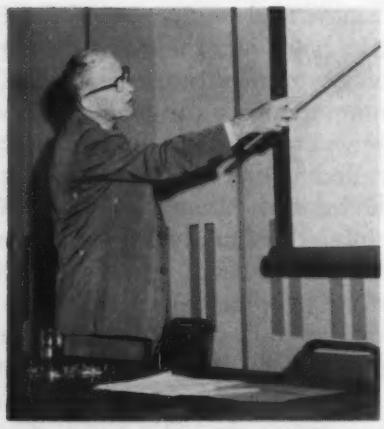
Curves of temperature rise as related to entering temperature were plotted for all combinations of duct diameter, insulation thickness, duct air velocity, vapor barrier emissivity and ambient temperatures, and were presented.

It was pointed out that these data were based upon the assumption that condensation does not occur on the duct or insulation surfaces. For uninsulated ducts, the surface temperature was indicated as a function of the duct air temperature, ambient temperature and duct air velocity.

It was found that the vapor barrier surface temperatures for insulated ducts was practically independent of duct air temperature and velocity within the range investigated; it was principally dependent upon ambient temperature, surface emissivity and insulation thickness. The surface temperatures were shown as a function of ambient temperature with bands for each vapor barrier emissivity encompassing all variations due to duct air temperature and velocity.

Results of balloting by the members in regard to the Palmatier resolution, the election of officers and approval of amendments to the By-laws were announced officially by Executive Secretary R. C. Cross

How the Navy approaches the estimating of energy use by projected central air conditioning plants was the topic of A. E. Congress



Clark Humphreys, ASHRAE's Assistant Director of Research, presented two Laboratory papers at the Third Technical Session on behalf of their authors (Schutrum-Ozisik and Kerka)

Clancy Pollock presided at the Second Technical Session, where refrigerants and other refrigeration system components were discussed

The paper of J. R. Wright covered calculated temperature rise in round ducts

The topic of W. P. Brown, at the Fourth Technical Session, was a hot box cold room facility

Morris Kaplan, Technical Director of the Consumers Union, keynoted the Domestic Refrigerator Engineering Symposium, where quality control was the topic and a large group heard talks by representatives of three leading manufacturers

The lower emissivity, in conjunction with reducing the temperature rise in ducts, caused a lower surface temperature.

The authors indicated that they were reporting on but one phase of a comprehensive investigation conducted in Warm Air Heating Research Residence No. 2 under the terms of a cooperative agreement between the National Warm Air Heating and Air Conditioning Association and the Engineering Experiment Station of the University of Illinois.

A special guarded hot box, to be used in conjunction with a large cold room for the measurement of the heat transmission coefficients of building sections 4 ft wide and 8 ft high, was described by W. P. Brown, K. R. Solvason and A. G. Wilson. Designed and built by the Division of Building Research of the National Research Council of Canada, a section to be tested is incorporated as part of a partition which separates the cold room into two compartments. The smaller compartment is heated electrically to maintain any desired constant warm side temperature from 65 to 75 F; the larger compartment is refrigerated to maintain the desired constant cold side temperature from 40 to –60 F. It was indicated that the guarded hot box is positioned against the warm side of the building section and is heated electrically to maintain a constant temperature. The heat transmission coefficient is then calculated from the measured electrical input and the temperatures.

The authors reported that heat flow through any 4 by 8-ft section of test walls up to 12 ft in length can be metered reliably with this facility. The total error in determination of thermal transmittance coefficients depends upon test conditions and on wall thermal conductance. For a wall of U value 0.1 Btu/hr/sq ft/F at a temperature difference of 30 F the total of panel heat leakage, control and measurement errors, if additive, is 2.8%. It was indicated that the edge heat leakage error depends upon wall construction and edge arrangement. For thin wall sections with the edge configuration used to date it is about 3%.

The error in measured thermal transmittance coefficient for such walls was cited as being between 0.2 and 5.8%. Alternative methods of sealing the box to the wall specimen which will largely eliminate edge heat leakage error are being considered, it was stated. The total percentage error will decrease as the temperature difference between hot box and cold room increases or as the thermal conductance of the wall increases.

It was pointed out that the error in measuring thermal conductance depends on the reliability with which an appropriate average surface temperature can be measured or estimated. Operation has indicated that heat transmission coefficients can be reproduced within 1.0%.

Since the temperature of the inside surface of the hot box is under direct control, it is possible to operate the box under conditions of natural convection without excessive vertical air temperature gradients and to exercise some control over the relationships between box surface and air temperature. It was indicated that with energy supplied only to the pump motor, the air temperature has been generally within 0.5 F of the box surface temperature in tests on typical walls.

"Air conditioning service of the required integrity at the lowest total annual cost – that is our objective," stated A. Eugene Congress, who observed that when various choices are possible, it is sometimes difficult to decide which system design, granting equally acceptable integrity, will result in the lowest total annual cost. The author described how such a problem was approached and the results obtained in the selection of the refrigeration plant to air condition a $5 million addition to a large naval hospital. The initial capacity required was designated as 900 ton with provision for future expansion to 2700 ton.

It was revealed that preliminary studies showed that a sizable capital investment would be required to provide electrical power capacity for an all-electric refrigeration plant. However, steam of sufficient capacity to supply the air conditioning was available year round from the existing central heating plant; the building layout made it possible for steam to be brought into the refrigeration building merely by punching a hole in a wall of the adjacent heating plant. It was pointed out that since the ultimate plant will contain multiple units, no stand-by capacity was considered necessary. Five different schemes were evaluated within the paper.

Two aspects of the problem which required special study were brought out: (1) determination of the initial capital investment and (2) estimating the annual operating cost. Estimates of procurement and installation costs were not difficult to obtain, but comparative estimates of operating costs – especially energy costs – presented a problem. It was pointed out that preliminary estimates assumed equivalent full-load operating hours for the refrigeration machine and total hours operation for the auxiliaries, but this approach was not found to be sufficiently accurate. It was pointed out that the major factor in evaluating relative annual operating costs is the energy charge.

The author reported that in his specific installation, the capital worth of annual savings justified the selection of a steam system. The question of maintenance was considered and it

was determined that in this project the difference in maintenance for the various schemes was negligible.

It was conceded, however, that this selection, while the most economical and logical under the conditions for this installation, will not necessarily be the same for other applications with different governing conditions. It was emphasized that a great deal depends upon the factors of availability and type of power; the incremental cost of fuel to make steam, electricity and water, the steam rate of selected turbines; profitable use of by-product steam and heat of condensate; applicable taxes, interest and insurance on capital investment.

Author Congress expressed the hope that the procedures demonstrated would be helpful in providing evaluations for service of the required integrity at the lowest total annual cost.

At the Industrial Ventilation Symposium – With Robert Jorgensen as Chairman, this session covered various aspects of Laboratory Hood Design. J. C. Burke, Jr., who discussed basic requirements of laboratory hoods, outlined criteria for toxic fumes, objectionable odors and other significant factors and the economics of hood design. E. J. Williams spoke on the design of hoods which incorporate a supply of unconditioned air down the face of the hood and the economy of their use in air conditioned laboratories. Construction features of hoods and various materials were described by T. B. Lanahan. Concluding this meeting, G. T. Saunders and L. N. Nelson presented a history of the use of fume hoods.

At the Air Conditioning Symposium – Under the direction of W. R. Moll, this session examined a controversial problem of present-day design – Should window units or a central system be used for residential applications?

At the Food Refrigeration Symposium – Guided by Aaron L. Brody, this Symposium explored extensively the use of liquefied gases for low temperature food handling. Chairman Brody noted the recent entry into the field of food refrigeration of the use of liquefied gases as cooling and blanketing agents for food handling; accelerated by the availability of large quantities of relatively inexpensive liquefied and solidified gases. At this meeting, experts from many phases of the industry, government and the academic world discussed various aspects, both pro and con, of the use of liquefied gases.

According to Kirby M. Hayes, adequate temperature protection for frozen foods is presently lacking in certain areas. Liquefied gases appear to be economical and versatile enough to offer solutions to many of these problems, it was indicated, and quality maintenance through temperature control can be provided with liquefied gas systems. Author Hayes expressed the opinion that although refinements in the systems will be made, liquefied gases will have an important future role in the frozen food field.

Relative merits of various gas liquefying cycles for use in freezing and transportation of foods were discussed by Samuel C. Collins. It was pointed out that in the freezing of food by contact with liquid nitrogen or other liquefied gases, it is generally inconvenient or impossible to use directly the refrigerative power of the cold vapor; the author covered various methods of avoiding this loss.

The complete freezing of biological materials at the ultra-low temperatures of liquid gases presents problems not yet answered by science, relative to their mode of handling in the marketing channels, observed Walter A. Maclinn. Reviewing information pertaining to the freezing of tissues as seen at the molecular level, Author Maclinn spoke of the gaps of knowledge regarding the handling of ultra-low temperature frozen foods.

The problems inherent in multi-stop delivery of frozen foods using liquid nitrogen refrigeration were outlined by J. J. Kane, who stated that the stop and go nature of the operation precludes the possibility of the refrigeration system maintaining 0 F environment. The author described a system using liquid nitrogen as the primary refrigerant, which is currently in commercial use.

Truck and rail car pre-cooling and post-cooling with carbon dioxide was the topic of John P. Antolak. It was indicated that carbon dioxide was found to be beneficial in pre-cooling and post-cooling equipment used in transporting perishable foods, whereby the carbon dioxide is injected into the car or trailer to supply a high refrigeration capacity to quickly reduce temperatures to desirable levels. Various forms of carbon dioxide and methods of use were discussed.

In his presentation, Robert C. Webster covered the use of liquefied gases for freezing and shipping of goods. Commentaries were offered by V. J. Johnson and Irving J. Pflug.

At the Domestic Refrigerator Engineering Symposium – According to Chairman E. J. Von Arb, product quality has become more than just desirable – it is a necessity and achieving it has become the first requisite of effective corporate leadership. This Symposium related product quality, quality control and product reliability. The keynote address was by Morris Kaplan.

(*Continued on page* 92)

Thermodynamic investigation of a *Refrigerant Expansion Engine*

THOMAS M. OLCOTT

HAROLD A. BLUM

The principle of substituting a refrigerant expansion engine for the expansion valve of a vapor compression cycle system is well known and correspondingly documented. Published performance data for such an engine, however, are conspicuously lacking.

This investigation consisted of (1) construction of a simple engine, (2) testing the engine and (3) comparison of its performance with the isentropic expansion.

With this device, the recovery of available energy was only as high as 10%. The present analysis shows that the pressure drop and leakage in the valves were major causes of the low energy recovery. If these were eliminated, the percentage recovery could have reached 40%.

A theoretical comparison was made between the single circuit refrigeration system, utilizing the expansion engine and the cascade system.

THERMODYNAMIC BACKGROUND

An ideal vapor compression cycle refrigeration system departs from a Carnot refrigeration cycle in three ways. This paper discusses these differences and presents a device that will allow the ideal vapor compression cycle to partially recover some of its losses.

Both the ideal vapor compression cycle and the reversed Carnot cycle are shown on a temperature-entropy diagram (Fig. 1) and on a pressure-enthalpy diagram (Fig. 2). The ideal vapor compression cycle is shown by the points 1, 2, 3, 4, 1, 1. It is a constant enthalpy expansion from points 1 to 2, followed by a constant pressure and temperature vaporization to point 3, then an isentropic compression to point 4, and finally a constant pressure cooling and condensation back to point 1. The reversed Carnot cycle follows the path shown by points 1, b, 3, c, 1. This cycle consists of an isentropic expansion and compression from points 1 to b and from points 3 to c. It also includes a constant temperature condensation and vaporization from points c to d and from points b to 3 respectively.

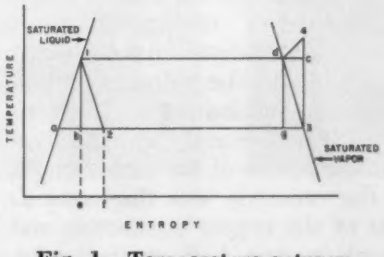


Fig. 1 Temperature-entropy diagram

These two cycles differ in the following manner. First, the vapor compression cycle system includes the area d, c, 4, d, known as the superheat horn. The inclusion of this area causes an increase in both the heat rejected in the condenser and the work required for compression. The vapor compression cycle also fails to recover work in the expansion from points 1 to 2. This is represented by the area b, c, f, 2, b, which is equal to the enthalpy change from point 2 to point b and is therefore equal to the work lost for an adiabatic process. Finally, the vapor cycle suffers a loss of cooling due to the increase in entropy during the throttling expansion. This loss is also shown by the area b, e, f, 2, b on the temperature-entropy diagram.

There are two possible ways that the superheat horn can be removed. To have a constant temperature cooling of the superheated vapor from points c to d would be one method. This would be impossible, since it requires a rise in pressure and a decrease in volume with no input of work. Another way to remove the superheat horn might be to begin the isentropic compression at point g. This, however, requires compression through the wet region.

The remaining losses are those associated with the throttling expansion. These losses can be par-

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tially eliminated by allowing the refrigerant to expand against a piston, so that work may be removed from the expansion. If the output of this piston were usefully utilized, there would be both an increase in the net refrigeration effect and a reduction in the work that would have to be supplied. This cycle is represented by the points 1, b, 3, 4, d, 1 on Figs. 1 and 2.

The performance of refrigeration cycles is studied by comparing their coefficients of performance. The coefficient of performance (C.O.P.) is defined as the net refrigeration effect divided by the net work that must be performed on the cycle. The net refrigeration effect is equal to the enthalpy change during the constant pressure vaporization. The net work performed on the cycle is equal to the enthalpy change during the compression process minus any work that is produced by any other part of the cycle.

The coefficients of performance of the various cycles discussed are as follows: (See Fig 2)

The ideal vapor cycle with throttle expansion

$$\text{C.O.P.} = \frac{H_3 - H_2}{H_4 - H_2}$$

The reversed Carnot cycle

$$\text{C.O.P.} = \frac{H_3 - H_b}{(H_c - H_3) - (H_2 - H_b)}$$

The ideal vapor cycle with piston expansion

$$\text{C.O.P.} = \frac{(H_3 - H_2) + (H_3 - H_b)}{(H_4 - H_3) - (H_2 - H_b)} = \frac{H_3 - H_b}{(H_4 - H_3) - (H_2 - H_b)}$$

It is easy to see that the vapor compression cycle with piston expansion is different from the Carnot cycle only by the additional work of compression required from point c to point 4. In Table I the performance improvement of the vapor compression cycle with a piston expansion over a throttle expansion is shown, using the standard ton characteristics and with Refrigerant 12.

EXPERIMENTAL EQUIPMENT

As the building of an isentropic expansion device would be impossible, it was decided to produce a simple expansion engine and to determine how much of the available work from an isentropic expansion could be recovered.

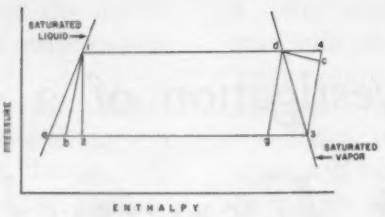


Fig. 2 Pressure-enthalpy diagram

The engine used was a small single cylinder refrigeration compressor. The valve plate was removed, and the inlet and discharge manifoldings were reamed out to reduce pressure drop. Also the passage leading from the crankcase to the inlet manifold was sealed, and provisions were made to vent the crankcase to a low pressure drum. Electrically operated solenoid valves were placed at the inlet and discharge of the compressor. A wooden fly wheel (eight in. diam) was fastened to the compressor shaft. Strips of linoleum were fastened to the outer edge of the fly wheel, which activated micro-switches, which in turn operated the solenoid valves.

By changing the length of these strips, both the inlet and discharge valves' timing could be varied. Also fastened to the wooden fly wheel was the reducing motion for the engine indicator. This consisted of a slide and crank mechanism. The ratio of the crank length to the eccentric was the same as that of the engine connecting rod length to one half of its stroke. Therefore, the indicator motion was directly proportional to the piston travel at every point. The engine indicator used was a standard Crosby Steam Engine Indicator. Figs. 3, 4 and 5 show a top, front, and back view of the engine.

Table I Standard Ton of Refrigerant 12

		Throttle Expansion	Piston Expansion
Evaporator Temperature	deg F	5	5
Condenser Temperature	deg F	86	86
Net Refrigeration Effect	Btu/lb	50.035	51.633
Work of Compression	Btu/lb	10.636	9.038
Coefficient of Performance	Btu/lb	4.704	5.713

The engine cylinder and crankcase, as well as the inlet and discharge lines, were well insulated. Both the inlet and the discharge lines were instrumented with pressure gauges. The crankcase of the engine was vented to the discharge line to allow any refrigerant that might leak past the piston to flow to the low pressure drum. This prevented any pressure build up in the crankcase.

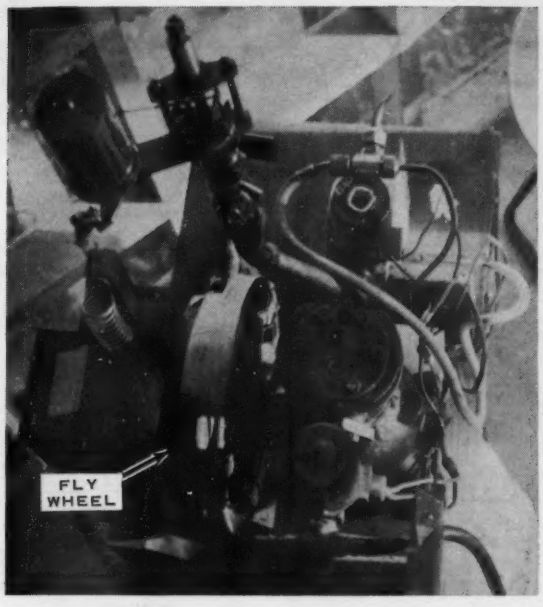
The engine was operated by flowing high pressure liquid Refrigerant 12 through it and then allowing the refrigerant to expand to a low pressure source. The high pressure liquid refrigerant was kept in a drum at approximately room temperature. This drum was inverted and fastened to the inlet valve. Thus it was insured that all of the refrigerant entering the engine would be liquid. After the refrigerant expanded in the engine, it was discharged to a second drum, which was packed in dry ice.

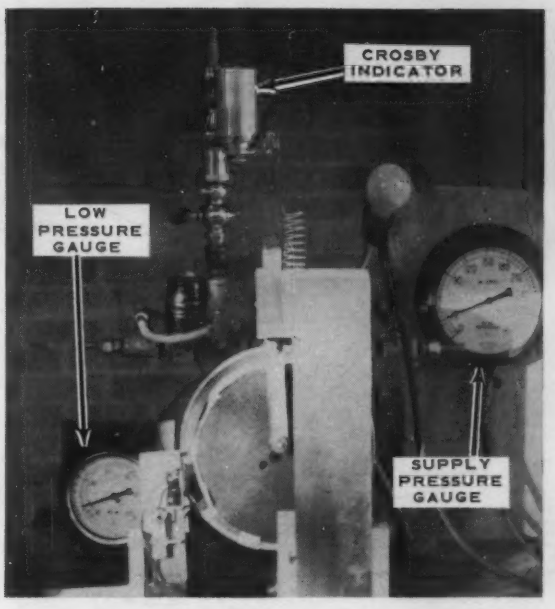
Mass flow of the refrigerant was determined by weighing the drums and observing the time of the run. Indicator cards were taken every 30 sec and engine speed was taken continuously by a direct count. This was possible since the engine speed was always about 60 rpm. The pressure in the supply drum, as well as the discharge drum, also was recorded during the run. Approximately 10 lb of refrigerant were used for each of the runs. This resulted in a run of approximately four min in length. Fig. 6 shows the complete test setup.

RESULTS

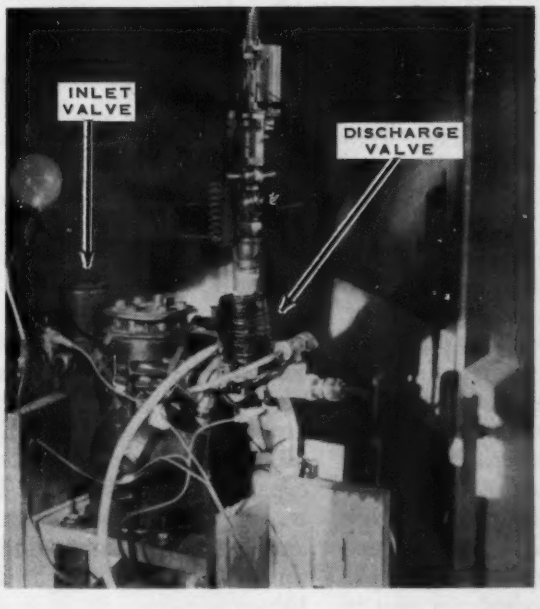
Ten satisfactory data runs were made during the testing of the engine. The data from these runs are presented in Table II.

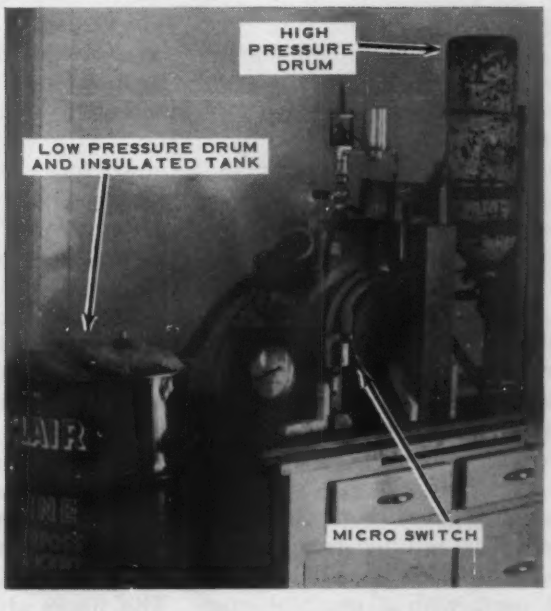
The three inlet valve angles that were used were 11.4, 13.9, and 17.6 past top dead center. The supply pressure varied between 80 and 142 psia, and the discharge pressure varied between 25 and 33 psia. The mass flow of refrigerant through the engine was nearly constant, varying between 2.1 and 2.4 lb per min. The speed of the engine varied between 45 and 71 rpm. The





Figs. 3, 4, 5, 6 Top, front and rear views of the engine and of the entire test facility





indicated horsepower taken from the indicator cards varied between 3.39×10^{-3} and 5.74×10^{-3}. This, divided by the mass flow and converted to the proper units, gave the enthalpy change across the engine, which varied between 0.0621 and 0.0980 Btu per lb.

Typical indicator cards from each of the ten runs are shown in Figs. 7 and 8. Included on these cards are their areas and the ambient pressure line. Cards were taken every 30 sec during each run, which resulted in 5 to 8 cards for each run. The deviation in area from card to card for a given run was never more than 20%.

DISCUSSION OF RESULTS

The mean effective pressure was taken from all of the indicator cards during each of the ten runs. It was then averaged and combined with the average engine speed for that run to determine the indicated horsepower. The engine constant was 5.82×10^{-6}, which came from a 1-7/16-in. bore and stroke. The engine constant multiplied by the mean effective pressure and engine speed yields indicated horsepower. The spring constant for the Crosby Indicator was 20 psi per in. for all runs. The mass flow of refrigerant was determined by taking the loss of weight of the supply drum and dividing it by the time length of the run. The supply and discharge pressures were taken directly from the calibrated gauges in the supply and discharge lines.

During the testing the inlet valve timing and the pressure in the supply drum were varied. Several runs were made at each of the three inlet valve openings. From examination of the data it appeared that the variation in the inlet valve opening, in this range, had no effect on the performance of the engine.

The indicated power correlated well with the supply pressure

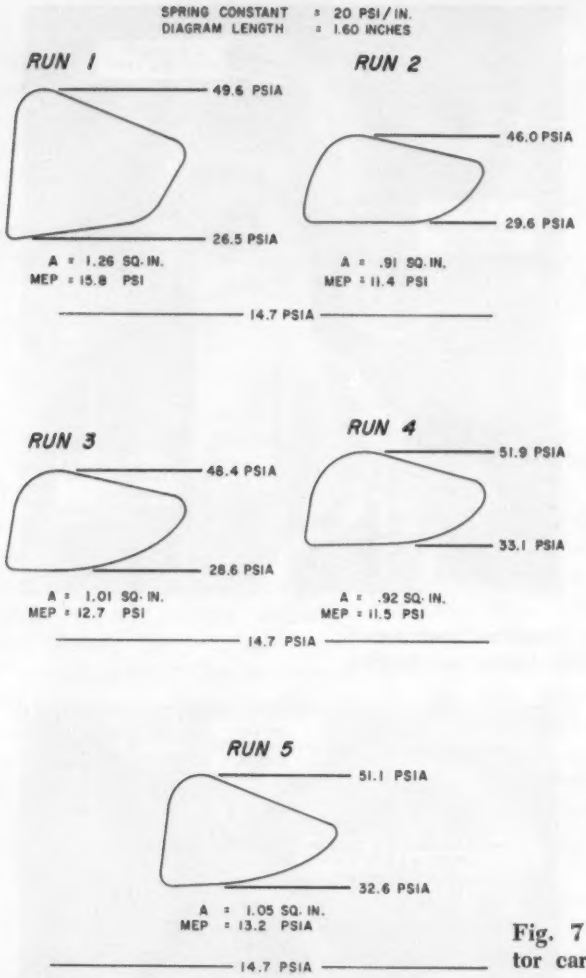


Fig. 7 Typical indicator card from each run

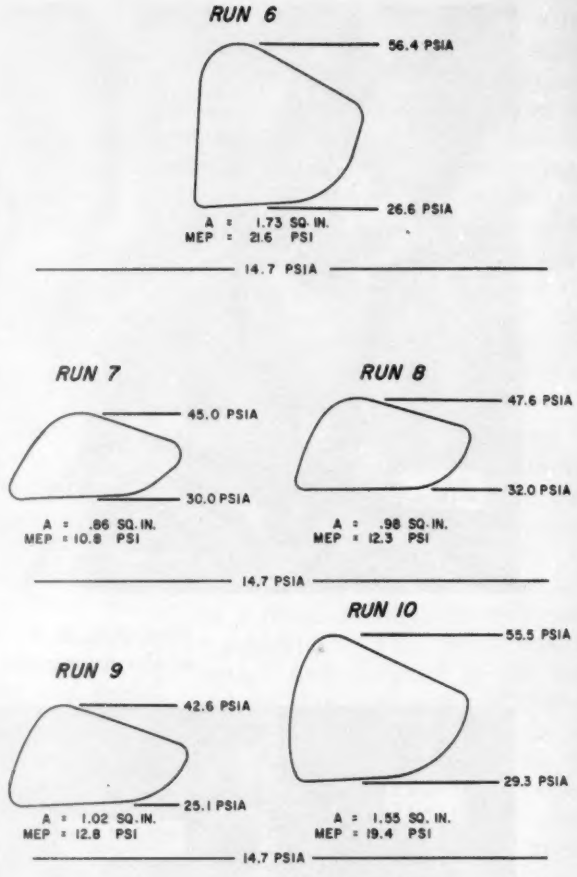


Fig. 8 Indicator cards

to the engine. These data are shown in Fig. 9 and reveal an increase in indicated power with increasing supply pressure. The enthalpy loss across the engine, which was plotted in Fig. 10, varied in the same manner as the indicated power. This resulted from the engine having been operated, for the most part, with a constant mass flow. The enthalpy loss across the engine was equal to the indicated power divided by the mass flow, since the engine was well insulated to prevent any heat loss.

The engine had no external load applied during any run, thus making the brake power always equal to zero. Consequently, the indicated power was equal to the friction power.

It is interesting to compare the work extracted from the refrigerant by this engine with the work that could have been extracted by performing an isentropic expansion between the engine's supply and discharge pressures. The work realized from such an isentropic expansion would be the maximum possible. This maximum work would also be equal to the maximum enthalpy change that the refrigerant could have. Fig. 11 shows a plot of the enthalpy change of the refrigerant across the engine versus the minimum enthalpy change that could have occurred if the expansion were isentropic. This figure also includes a plot of the engine efficiency, or percent of the maximum available enthalpy change, that was recovered by the engine. This curve is just the slope of the previous curve. This efficiency varied between 11 and 4%.

FACTORS AFFECTING PERFORMANCE

Reasons for low reported efficiencies can be seen best by studying the process that occurred in the engine during a complete stroke. Fig. 12 is a plot on a pressure-enthalpy diagram showing the conditions of run number 7. Point 1 on this figure is the state of the saturated liquid refrigerant in the supply drum.

As the liquid passed through the inlet valve, there was a considerable pressure loss due to the high flow through the valve. This pressure loss was pure throttling and is shown as a constant enthalpy expansion to point 2. The first reason why the efficiency of this engine was so low is this large throttling loss through the inlet valve.

The pressure at point 2 was determined by taking the average maximum pressure from the indicator cards. The measured pressure loss agrees with the calculated value for this size valve and flow conditions. A calculation of this loss for run number 7 is presented in part A of the Appendix. This calculation shows that with a flow of 1.24 lb per sec through the valve there will be a loss of 50 psi across the valve. This agrees with the measured flow of 1.28 lb per sec and the pressure loss of 50 psi.

Point 2 is the maximum pressure in the engine and represents the beginning of the engine expansion. This expansion continues to point 3, which indicates the end

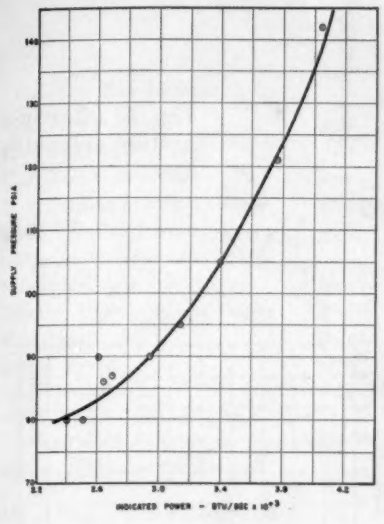


Fig. 9 Indicated power vs. supply pressure

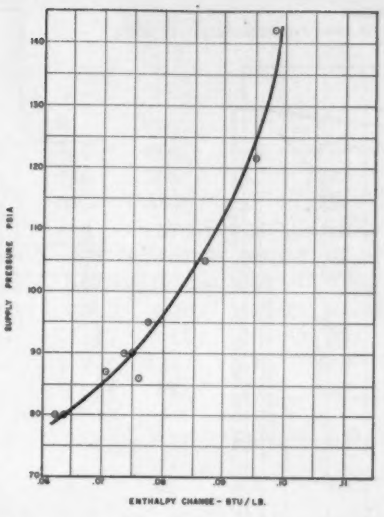


Fig. 10 Enthalpy loss across engine

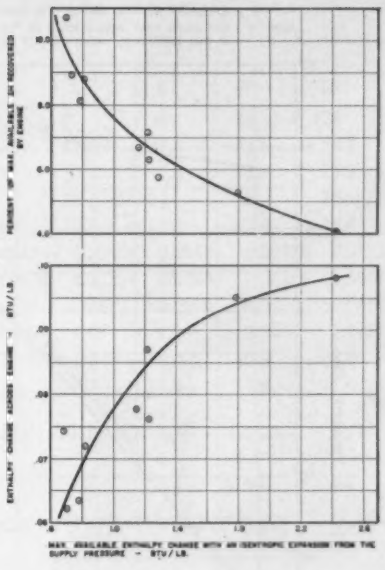


Fig. 11 Engine efficiency including all losses

of the discharge stroke where the pressure in the engine is the least. The constant enthalpy expansion from point 3 to point 4 is the loss across the discharge valve.

During the expansion from point 2 to point 3 not only is enthalpy being removed due to the work that is being done, but also some heat transfer is taking place between the engine and the refrigerant.

If this test had been performed by perfectly insulating the engine from its surroundings and by then allowing the refrigerant to flow through the engine until equilibrium conditions were reached, the net contribution of any heat transfer effects would have been zero. Thus, the engine would have maintained an average temperature that would have been between the temperature at point 2 and the temperature at point 3. During the first part of the power stroke some heat would have been transferred from the engine to the refrigerant. As the refrigerant cooled to a temperature below that of the engine, the same quantity of heat would have been returned to the engine. Thus the net enthalpy loss of the refrigerant would have been equal to the work removed.

The reason that this test was not conducted in the manner just described was that the quantity of refrigerant required to cool the engine to equilibrium conditions was a good portion of the amount available for the entire run. Also a continuous method of determining the mass flow of refrigerant would have been required since the testing would have had to begin when equilibrium conditions were reached.

In the preliminary runs while the engine was being prepared, it was determined that the pressure at the beginning of the expansion was approximately 45 psia and at the end of the expansion approximately 30 psia with the corresponding saturation temperatures of 30 and 12 F respectively. Therefore, in an attempt to bring the engine to equilibrium conditions before the test began, the engine was insulated with glass wool, and then a few pieces of dry ice were placed near the cylinder to cool it to approximately 15 to 20 F. This temperature was noted by observing the pressure of the saturated gas in the engine crankcase.

The clearance volume of this engine is 6.3 cu in. and the swept volume is 2.3 cu in. This means that the swept volume is only 27% of the total volume. This percentage is quite low, as in most engines the swept volume is several hundred percent of the clearance volume.

The clearance volume multiplied by the change in density from point 2 to point 3 (Fig. 12) should be equal to the mass of refrigerant admitted during a complete stroke. This is true because both points 2 and 3 occur at top dead center, point 3 being just prior to the opening of the inlet valve and point 2 being just after the opening of the inlet valve. For run number 7 the density at point 2 is 6.76 lb per cu ft and at point 3 is 3.58 lb per cu ft. The density difference multiplied by the clearance volume yields a mass of 0.0116 lb. However, the mass admitted during each stroke in run number 7 was 0.035 lb; three times as much as would have been expected.

The reason for this huge discrepancy is the other major factor that caused the engine to have such a low efficiency. As was previously stated, the valves used on the engine were electrically operated solenoid valves. These were necessary, since the reed valves that were initially in the compressor did not operate in the proper manner for engine use. The refrigeration solenoid valve uses the power of the solenoid for opening and the valve's upstream pressure for closing. The valve is a servo mechanism during its closing operation, using its own upstream pressure as servo pressure.

The diagram in Fig. 13 is a sketch of how this type of valve operates. As can be seen from this sketch, the valve stem falls due to gravity when the solenoid coil is deenergized, and there is no difference between the inlet and exit pressure. However, in this condition the stem is poorly seated. When the inlet pressure is increased, the pressure on the valve

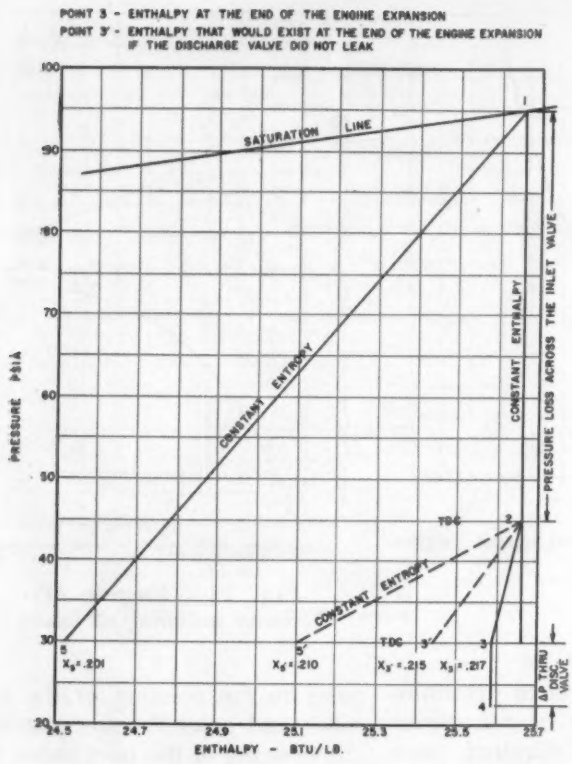

POINT 3 - ENTHALPY AT THE END OF THE ENGINE EXPANSION
POINT 3' - ENTHALPY THAT WOULD EXIST AT THE END OF THE ENGINE EXPANSION IF THE DISCHARGE VALVE DID NOT LEAK
SATURATION LINE
CONSTANT ENTROPY
CONSTANT ENTROPY
CONSTANT ENTHALPY
PRESSURE LOSS ACROSS THE INLET VALVE
ΔP THRU DISC VALVE
TDC
TDC
PRESSURE P.S.I.A
ENTHALPY - BTU/LB.
X_2 = .201
$X_{3'}$ = .210
X_3 = .215
X_3 = .217

Fig. 12 Conditions of expansion for run No. 7

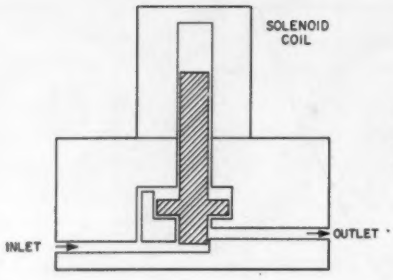


Fig. 13 Refrigeration solenoid valve

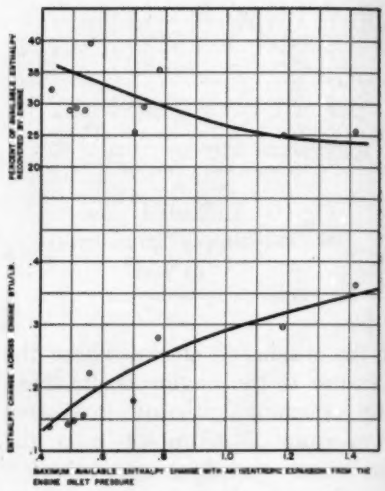


Fig. 14 Engine efficiency eliminating the effect of valve losses

seat tends to open the valve, and the pressure on top of the piston above the valve seat tends to close the valve. Since the area of the piston is greater than the valve seat, the net effect is to close the valve. This system works quite well for most operations.

However, when there is a sudden rise in the inlet pressure, a small amount of flow leaks past the valve seat. This is because the flow path to the piston is much smaller than that through the valve. In the case of this engine, the discharge valve leaked nearly two-thirds of the total flow admitted by the inlet valve before it closed tightly. This was observed when the engine was being run on air and was discharging to ambient. When the inlet valve opened, a small burst of air was passed by the discharge valve.

Three different types of solenoid valves were tried as discharge valves, but they all had this tendency. Also the discharge valve was closed 34 deg before the piston returned to top dead center in an attempt to seat the discharge valve during the compression part of the stroke. However, the pressure build up was not enough to cause the valve to close completely. This can be seen by looking at the indicator cards and by noting that there was no compression portion of the discharge stroke. Thus it appears that the exhaust valve closed at the same instant that the inlet valve opened. This effect caused the enthalpy change across the engine to be nearly one-third less than it could have been with better valves.

Of course some of this leakage flow was going past the piston; but since the compressor piston and cylinder walls seem to be in good condition, this leakage was probably quite small. Therefore the two major causes of inefficiency were brought about by the inlet and discharge valves. When these effects are eliminated from the experiment by considering an isentropic expansion from point 2 to point 3 (Fig. 12) using only the mass of refrigerant that remained in the cylinder during the power stroke, the percent of work recovered varies between 25 and 40%. These data are shown in Fig. 14 and the method of calculation is described in part B of the Appendix for run number 7.

Comparison between piston expansion, cascade operation and multi-stage compression – In low temperature refrigeration systems cascade cycles commonly are used to improve the systems' coefficient of performance. In the cascade system a series of refrigerants with progressively lower boiling points is used in a series of single stage units. The evaporator of the first system, operating at the highest temperature, is used to cool the condenser of the second system; this can be continued for as many cycles as desired. Thus each refrigerant circuit is a system in itself, and each refrigerant can be chosen so that it operates best within the required temperature and pressure range.

Another cycle commonly used to improve C.O.P. is multi-stage compression with both intercooling and subcooling. In this cycle a portion of the liquid leaving the condenser is subcooled by partial vaporization of the remaining liquid. The subcooled portion then passes to the evaporator and on to the first compressor. At the discharge of this compressor the two refrigerant circuits join in the flash intercooler and then pass through the second compressor and on to the condenser.

Table II Test Data

Run Number		1	2	3	4	5	6	7	8	9	10
P_1 = Supply Pressure	psia	90	80	80	90	87	142	95	105	86	121
P_2 = Max. Press in Engine	psia	49.6	46.0	48.4	51.9	51.1	56.4	45.0	47.6	42.6	55.5
P^3 = Discharge Pressure	psia	26.5	29.6	28.6	33.1	32.6	26.6	30.2	32.0	25.1	29.3
P_4 = Press. in Low Temp. Drum	psia	15	20	20	25	25	15	20	23	15	20
Mean Effective Pressure	psi	15.8	11.4	12.7	11.5	13.2	21.6	10.8	12.3	12.8	19.4
Indicated Horsepower	hp	0.00413	0.00353	0.00339	0.00368	0.00382	0.00574	0.00444	0.00481	0.00372	0.00530
Indicated Power	Btu/Sec	0.00294	0.00251	0.00240	0.00261	0.00270	0.00405	0.00314	0.00340	0.00264	0.00376
Enthalpy Change Across Engine	Btu/lb	0.0725	0.0632	0.0621	0.0742	0.0702	0.0980	0.0770	0.0865	0.0760	0.0950
Mass Flow	lb/min	2.27	2.38	2.32	2.11	2.31	2.44	2.45	2.36	2.08	2.37
Engine Speed	rpm	45.0	53.0	46.0	55.0	50.0	45.5	71.0	68.0	50.0	47.0
Inlet Valve Angle	deg	13.9	17.6	17.6	17.6	13.9	13.9	11.4	11.4	11.4	11.4
ΔH Available, expanding isentropically from point 1	Btu/lb	1.28	0.78	0.71	0.69	0.83	2.42	1.16	1.21	1.22	1.79
Engine Efficiency, including all losses	%	5.7	8.1	8.8	10.7	8.5	4.1	6.7	7.2	6.2	5.3
ΔH of Engine, eliminating valve losses	Btu/lb	0.179	0.149	0.139	0.142	0.156	0.364	0.222	0.277	0.215	0.297
ΔH available with an isentropic expansion eliminating valve losses	Btu/lb	0.70	0.51	0.43	0.49	0.54	1.42	0.56	0.78	0.73	1.18
Engine Efficiency, eliminating valve losses	%	25.5	29.3	32.3	29.0	29.0	25.7	39.9	35.4	29.5	25.0

Table III presents a comparison among the 1) cascade system, 2) multi-stage compressor system, 3) single-stage system with throttle expansion and 4) single-stage system with piston expansion. The load is absorbed at −80 F and rejected at 80 F in each system. Several different refrigerants also are included in the comparison. It can be seen from this table that the single-stage system with piston expansion has the best performance when the expansion efficiency is as high as 75%.

The comparison of these four systems was computed assuming compression to be adiabatic (Appendix, Part C). It should be noted that the corrections for compression efficiency would affect the single stage compression units more unfavorably than the multi-stage compression units.

FUTURE DEVELOPMENT

While the engine described in this paper successfully demonstrates the principle of the refrigerant engine, the fraction of the isentropic process available energy actually recovered was too small to be commercially interesting.

To make this system a contribution to the art, this principle should be applied to an operating refrigeration cycle in such a way that work obtained from the engine be used either to reduce the work input to the compressor or handle auxiliary equipment.

APPENDIX

A. The following is an estimate of the pressure drop across the inlet valve to the engine for the conditions of run number 7. This is the pressure drop calculation method outlined by the ASRE DATA BOOK, Refrigeration Applications, 6th Edition, 1949.

Mass flow of refrigerant = 0.0345 lb/rev

Length of time the inlet valve was open = 0.027 sec/rev

Mass flow through the inlet valve = 1.28 lb/sec

Equivalent length of a 3/16-in. orifices solenoid valve = 15 ft of 3/16-in. straight pipe

Flow giving 1-psi pressure drop per 100 linear ft of 2-in. schedule 40 steel pipe = 21,000 lb/hr of Refrigerant 12

Multiplier H_1 to be applied to 2-in. schedule 40 steel pipe to find the flow giving the same pressure drop in 3/16-in. pipe = 0.004

Measured pressure drop across the inlet valve = 50 psi

Multiplier H_3 to be applied to the flow to give a pressure drop of 50 psi = 8.0

Therefore the flow that gives a 50-psi pressure drop in a 3/16-in. orifice solenoid valve is

$$\frac{(21{,}000)\ (0.004)\ (8)\ (100)}{(3600)\ (15)} =$$

1.24 lb/sec

B. The following calculation is a determination of the efficiency of the engine if the effect of the inlet valves' pressure loss and the discharge valve's leakage were eliminated. This calculation is done for run number 7

The supply pressure = 95 psia

The enthalpy of the saturated liquid entering the inlet valve = 25.667 Btu/lb

Therefore the enthalpy leaving the inlet valve = 25.667 Btu/lb

The pressure leaving the inlet valve = 45.0 psia

This yields a quality at the beginning of the expansion = 0.156

The density of the refrigerant at the beginning of the expansion = 6.76 lb/ft^3

The entropy of the refrigerant at the beginning of the expansion = 0.0546 Btu/lb F

Table III

System Type	Expansion	Expansion Efficiency	Refrigerant	C.O.P.
1) Cascade	Throttle	0%	R 22/R 12	1.76
Cascade	Throttle	0%	R 13/R 22	1.65
2) Multi-Stage	Throttle	0%	R 22	1.92
Multi-Stage	Throttle	0%	R 12	1.88
3) Single Stage	Throttle	0%	R 12	1.69
Single Stage	Throttle	0%	R 22	1.53
4) Single Stage	Piston	50%	R 12	1.79
Single Stage	Piston	75%	R 12	1.88
Single Stage	Piston	100%	R 12	1.97
Single Stage	Piston	50%	R 22	1.80
Single Stage	Piston	75%	R 22	1.97
Single Stage	Piston	100%	R 22	2.15

The pressure at the end of the expansion = 30.2 psia

If the expansion were isontropic the quality at the end of the expansion = 0.210

This yields an enthalpy of 25.109 Btu/lb

Therefore the enthalpy change with an isontropic expansion = 25.667 — 25.109 = 0.553 Btu/lb

The clearance volume of the engine = 0.00365 ft^3

The clearance volume multiplied by the change in density from the beginning of the expansion to the end of the expansion is equal to the mass of refrigerant admitted in each stroke

This is true since the piston is at top dead center at the beginning and again at the end of the expansion stroke.

To determine the density at the end of the expansion would require knowing the quality of the refrigerant at the end of the expansion. However, only the work done by the refrigerant is known from the indicator cards.

For run number 7 this = 0.00265 Btu/rev

To determine the density and enthalpy at the end of the expansion a quality is assumed which must be verified.

Assume this quality = 0.210

Density at the end of the expansion = 3.57 lb/ft^3

The density difference multiplied by the clearance volume yields a mass of refrigerant = 0.0116 lb/rev

This mass divided into the work done by the refrigerant yields an enthalpy change = 0.229 Btu/lb

Therefore, the enthalpy at the end of the expansion = 25.438 Btu/lb

However, this enthalpy occurs at a quality 0.2145

The density at this quality = 3.47 lb/ft^3

Recalculating the enthalpy change from this density = 0.222 Btu/lb

The enthalpy = 25.445 Btu/lb

This enthalpy occurs at a quality = 0.215

This is very close to value used

Thus, the engine efficiency $= \frac{0.222}{0.558}$

$= 39.9\%$

C. The following is a calculation of the coefficients of performance of (1) a cascade system, (2) a multi-stage compression system, (3) a single-stage system with throttle expansion and (4) a single-stage system with piston expansion. All four systems are designed to supply refrigeration at an evaporator temperature of —80 F and to reject heat at a condenser temperature of 80 F.

In the second cascade system the load at —80 F is absorbed by a Refrigerant 22 unit with an 80 F condenser. The Refrigerant 13 condenser is —10 F and the Refrigerant 22 evaporator is —20 F.

Heat absorbed in the R 13 evaporator = (50.6 — 7.5) = 43.1 Btu/lb R 13

Heat rejected in the R 13 condenser = (61.0 — 7.5) = 53.5 Btu/lb R 13

Heat absorbed in the R 22 evaporator = (102.8 — 34.3) = 68.5 Btu/lb R 22

Work of compression for the R 13 unit = (61.0 — 50.6) = 10.4 Btu/lb R 13

Work of compression for the R 22 unit = (123 — 102.8) = 20.2 Btu/lb R 22

Ratio of the mass flow of R 22 to R 13

$= \frac{53.5}{68.5} = .78$ lb R 22/lb R 13

$$\text{C. O. P.} = \frac{43.1}{10.44 + (20.2)(.78)} = 1.65$$

2. In the first two-stage unit with a subcooler and a flash intercooler the intermediate pressure is 27.4 psia, the evaporator —80 F and the condenser 80 F. Refrigerant 22 is the refrigerant. The liquid refrigerant is subcooled to the saturated intermediate pressure before entering the evaporator and the first stage compressor discharge is desuperheated in the flash intercooler.

Heat absorbed in the evaporator = 95.68 — 6.4) = 89.3 Btu/lb

Mass flow through the evaporator for a 1 ton load $= \frac{200}{89.3}$

$= 2.24$ lb/min

Combining the flash cooler enthalpy balance and subcooler enthalpy balance results in an enthalpy of the refrigerant leaving the subcooler and entering the intercooler of 86.0 Btu/lb. This results in a total mass flow through the high stage compressor of 3.45 lb/min. The work of compression for the low pressure stage unit = (80.5 — 68.5) (3.16) = 37.9 Btu/min

The work of compression for the high pressure stage unit = (89.5 — 75.5) (4.85) = 68.0 Btu/min

Thus:

$$\text{C. O. P.} = \frac{200}{106} = 1.88$$

3. For a single stage Refrigerant 12 unit with a —80 F evaporator and an 80 F condenser with throttle expansion

Heat Absorbed in evaporator = (68.47 — 23.37) = 45.10 Btu/lb

Work of compression = (96.01 — 68.47) = 27.54 Btu/lb

$$\text{C. O. P.} = \frac{45.10}{27.54} = 1.63$$

A single stage Refrigerant 22 unit with a —80 F evaporator and 80 F condenser with throtle expansion

Heat absorbed in the evaporator = (95.7 — 34.3) = 61.4 Btu/lb

Work of compression = (136 — 95.7) = 40.3 Btu/lb

$$\text{C. O. P.} = \frac{61.4}{40.3} = 1.53$$

4. A single stage Refrigerant 12 unit with piston expansion in lieu of throttling expansion and a —80 F evaporator and an 80 F condenser.

Quality of refrigerant entering the evaporator

$$= \frac{0.05475 + 0.02086}{0.20229} = 0.373$$

Enthalpy of refrigerant entering the evaporator

= —8.345 + (.373) (76.312) = 20.35 Btu/lb

Work performed by the expansion engine if the engine is 100 per cent efficient

= (23.37 — 20.35) = 3.02 Btu/lb

$$\text{C. O. P.} = \frac{45.10 + 3.02}{27.54 - 3.02} = 1.97$$

Work performed by the expansion engine if the engine is 75 per cent efficient

= (0.75) (23.37 — 20.35) = 2.26 Btu/lb

$$\text{C. O. P.} = \frac{45.10 + 2.26}{27.54 - 2.26} = 1.88$$

Work performed by the expansion engine if the engine is 50 per cent efficient

= (0.50) (23.37 — 20.35) = 1.51 Btu/lb

$$\text{C. O. P.} = \frac{45.10 + 1.51}{27.54 - 1.51} = 1.79$$

A single stage Refrigerant 22 unit with piston expansion, a —80 F evaporator and an 80 F condenser.

Quality of refrigerant entering the evaporator

$$= \frac{0.0708 + 0.0255}{0.280} = 0.344$$

Enthalpy of refrigerant entering the evaporator

= —10.22 + (0.344) (105.9) = 26.3 Btu/lb

Work performed by the engine if the engine is 100 per cent efficient

= (34.3 — 26.3) = 8.0 Btu/lb

$$\text{C. O. P.} = \frac{61.4 + 8.0}{40.3 - 8.0} = 2.15$$

Work performed by the engine if the engine is 75 per cent efficient

= (0.75) (34 — 26.3) = 6.0 Btu/lb

$$\text{C. O. P.} = \frac{61.4 + 6.0}{40.3 - 6.0} = 1.97$$

Work performed by the engine if the engine is 50 per cent efficient

= (0.50) (34.3 — 26.3) = 4.0 Btu/lb

$$\text{C. O. P.} = \frac{61.4 + 4.0}{40.3 - 4.0} = 1.80$$

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Providing an effective thermal barrier with

Transpiration Cooling

JOHN G. KRISILAS

JOHN E. BOBERG

Supersonic flight has evoked much interest in the aircraft industry during the past decade. Recently, added impetus was given by the advent of the B-70 bomber program and publication of various commercial supersonic transport studies. This paper deals with one major problem area common to high Mach number vehicles – the necessity of providing a highly effective thermal barrier to protect the interior crew and passenger compartments and equipment areas from the effects of high exterior skin temperatures arising from aerodynamic heating.

Rapid rise of exterior skin temperatures with Mach number is an established fact. For a typical case, exterior skin temperatures for a Mach 3 cruise flight at 80,000-ft altitude may be obtained as a function of exterior surface emissivity from Fig. 1. As can be seen, increasing the emissivity of the exterior skin, and thus its ability to radiate more heat to outer space, affords some relief from high skin temperatures. At the extreme of no solar heat being absorbed ($\alpha = 0$), and assuming perfect black body radiation ($\epsilon = 1$), a skin temperature in excess of 500 F will still be encountered. Thus, for supersonic flight, high skin temperatures will make mandatory provision of an effective thermal barrier.

THERMAL BARRIER CONFIGURATION

Illustrated in Fig. 2 is the basic scheme for the proposed thermal barrier. A layer of porous insulation (consisting of conventional glass fiber insulation batts) is located between the exterior skin and the interior trim, forming an air passage on either side. Cabin air is directed into the inner air passage and passed through the porous insulation in an outward direction to counteract the inward flow of heat. This is a specialized application of transpiration cooling, except that the cooling air is collected after passing through the insulation rather than continuing directly through the outer skin to the boundary layer. After passing through the insulation and being collected, the air may either be exhausted from the vehicle at a temperature approximately that of the exterior skin or it may be cooled by refrigeration equipment and recirculated. Therefore, the cooling potential of the air is utilized to its fullest in what corresponds to a highly effective heat exchanger.

ANALYTICAL MODEL

Differential equations and their solutions which govern heat trans-

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. . . by directing air through a porous insulating material counter to the direction of heat flow, a barrier to the heat flow is formed . . .

. . . the quantity of heat flowing into an insulated compartment may be reduced to a negligible amount by using a nominal thickness of insulation and moderate air flows . . .

. . . less heat would enter the aircraft interior as more air is expelled through the insulation . . .

. . . practical considerations limit the application to open or semi-open systems, where either all or a major portion of the transpired air is exhausted from the vehicle . . .

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fer and internal temperature distribution in a porous body undergoing transpiration cooling have been formulated to varying degrees of complexity by many investigators (references 1, 2, 3, 4 and 5). For the present investigation the most elementary model was assumed and, as will be shown, close agreement was achieved between the predicted temperature gradient and the measured gradient.

Analysis of transpiration cooled wall structures and evaluation of the effectiveness of this method of cooling is predicated upon being able to calculate the temperature distribution within the porous material and the rate of heat transfer through the wall structure for given temperatures on either surface. If surface temperatures are not known, then film coefficients must be evaluated both at the coolant-entry and the coolant-exit sides of the wall.

If negligible thermal resistance at the entry and exit boundaries is specified, reference 5 presents a method for calculating the solid material and coolant temperatures within the porous matrix of the wall structure. Because of the large area for heat transfer within the porous matrix between the coolant air and the solid material, the investigators cited in reference 5 found that the solid material and coolant temperatures become approximately equal at every point in the insulation. Some deviation was noted at the coolant-entry surface, but the temperatures became substantially equal a few pore diameters from this surface.

The present simplified analysis is based on a plane plate with unidirectional heat flow. Assumptions made were:

1. The coolant air and glass fiber material temperatures were equal at every point within the wall.

2. Mean apparent thermal conductivity of the glass fiber insulation measured at zero airflow was used in the equations.

3. Effects of porosity of the insulating material were neglected in the derivation.

4. Flow of the coolant is perpendicular to the wall at the entering side.

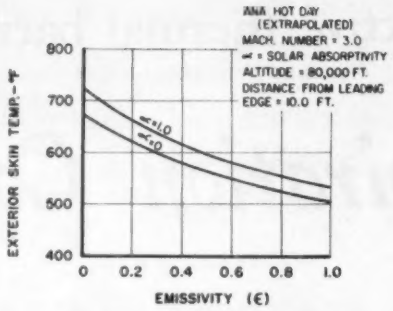


Fig. 1 External skin temperature as a function of exterior wall emissivity

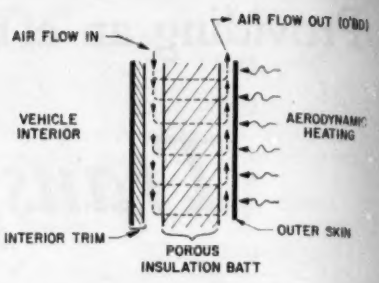


Fig. 2 Basic scheme for thermal barrier

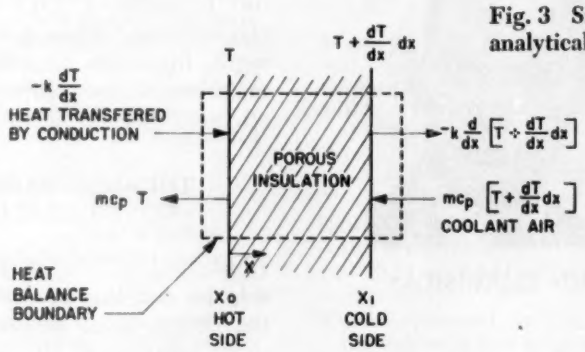


Fig. 3 Simplified analytical model

Shown in Fig. 3 is the simplified model on which the derivation was based.

Taking a heat balance through the dotted boundary

$$-k\frac{dT}{dx} + m\,c_p\left[T + \frac{dT}{dx}dx\right] = m\,c_p\,T - k\frac{d}{dx}\left[T + \frac{dT}{dx}dx\right]$$

where

- k = thermal conductivity of insulation
- T = temperature
- m = mass flow of air per unit of cross sectional area
- x = distance through insulation
- c_p = specific heat of air

The previous equation is simplified to

$$\frac{d^2T}{dx^2} + \frac{m\,c_p}{k}\frac{dT}{dx} = 0$$

or

$$\frac{d^2T}{dx^2} + B\frac{dT}{dx} = 0$$

where

$$B = \frac{m\,c_p}{k}$$

The solution to this linear differential equation is

$$T = C_1 + C_2\,e^{-Bx} \qquad (1)$$

The constants C_1 and C_2 are evaluated from the following boundary conditions:

at

$x = 0$ $\quad T = T_0$

$x = x_1$ $\quad T = T_1$

from which

$$C_1 = \frac{T_1 - T_0\,e^{-Bx_1}}{1 - e^{-Bx_1}}$$

$$C_2 = \frac{T_0 - T_1}{1 - e^{-Bx_1}}$$

The final equation for the temperature distribution then becomes:

$$T = \frac{T_1 - T_0\,e^{-Bx_1}}{1 - e^{-Bx_1}} + \left[\frac{T_0 - T_1}{1 - e^{-Bx_1}}\right]e^{-Bx} \qquad (2)$$

Heat transferred into the body from the hot surface, per unit surface area is:

$$q''_0 = -k\left[\frac{dT}{dx}\right]_{x=0} = -k\,(-B\,C_2)$$

$$q''_0 = \frac{k\,(B)\,(T_0 - T_1)}{1 - e^{-Bx_1}} \qquad (3)$$

Residual heat leaving the body of insulation at the cold surface and

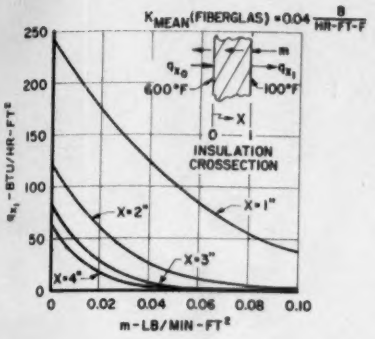


Fig. 4 Residual heat flow into cabin as related to transpiration air flow

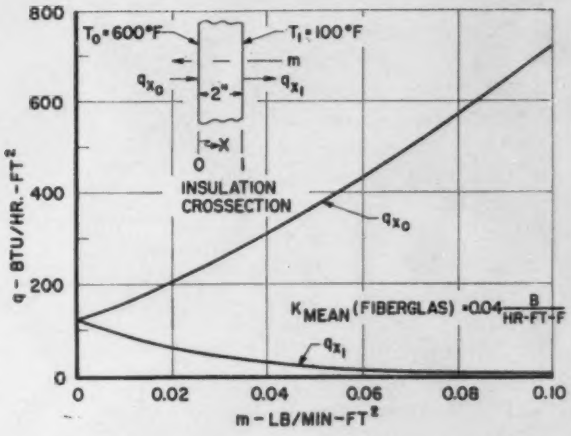


Fig. 5 Transpiration air flow effect on heat entering and leaving surfaces of insulation batt

not picked up by the outflowing air is:

$$q''_{x_1} = -k \left[\frac{dT}{dx} \right]_{x = x_1} = -k \left(-B\, C_2\, e^{-Bx_1} \right)$$

$$q''_{x_1} = \frac{k\,(B)\,(T_0 - T_1)\,e^{-Bx_1}}{(1 - e^{-Bx_1})} \qquad (4)$$

These equations permitted evaluation of the insulation scheme prior to the experimental program. For the present analytical and experimental study being considered, a typical design exterior skin temperature of 600 F was assumed. This is an average condition which requires, for the example cited of Mach 3 flight, a skin emissivity of 0.5 when all incident solar radiation is absorbed by the skin (see Fig. 1). A value of 100 F was chosen arbitrarily for the inner surface temperature.

Equation 4 may be utilized to evaluate the effect of varying the airflow through various thicknesses of insulation, and a solution to a typical problem is shown in Fig. 4. The amount of incoming heat not absorbed by the outward flowing air has been calculated for various air mass flow rates. It can be seen that the quantity of heat flowing into an insulated compartment may be reduced to a negligible amount by using a nominal thickness of insulation and moderate air flows. Comparison values for the usual case of zero air flow for various thicknesses of insulation may be obtained at the ordinate for m = 0.

It may be concluded that, for a given configuration, a small outward flow of air will reduce greatly the residual heat flowing into an insulated area. Conversely, for a given heat flow leaving the cold surface, much less insulation is required with the proposed insulation scheme.

Use of Equations 3 and 4 further illustrates the configuration characteristics. Fig. 5 shows the effect on the heat flowing into and out of a given layer of insulation with a constant imposed temperature gradient of 500 F. The curve labeled q_{x_1}, the heat leaving the cold surface of the insulation, was calculated from Equation 4; the heat entering, q_{x_0}, was calculated from Equation 3. Heat flow entering the insulation on the hot side increases as airflow is raised. The difference between the two curves represents the amount of heat absorbed by the outward flowing air. Heat flow through the insulation for zero airflow may also be obtained at m = 0. It is therefore apparent that the amount of heat absorbed by the air, even for moderate airflows, is greater than the heat input, using the same thickness of insulation but with zero airflow. Therefore, the proposed scheme is applicable to open systems, where air flowing through the insulation is exhausted from the vehicle. In some special cases it may be advisable to combine an open and closed system, whereby a portion of the air is collected, refrigerated and recirculated through the insulation while exhausting the remaining portion overboard.

EXPERIMENTAL APPARATUS

Figs. 6 and 7 show the test installation for determining the effect on heat transfer of air passing through glass fiber insulation. Means were provided to subject a nominal two-in. batt of insulation to a temperature difference of approximately 500 F while a small quantity of air was passed counter to the direction of heat flow. Comprising test equipment were the insulation test section, controllable radiant heat lamps, an air injector-type pump-

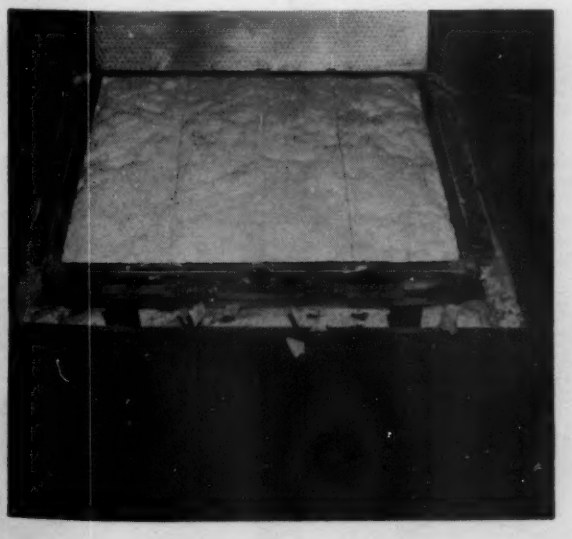


Fig. 6 Test installation

ing device and necessary temperature and airflow instrumentation, as shown in Fig. 7.

The test section consisted of a 0.020-in. aluminum frame which received four ½-in. insulation batts having a face area of four sq ft. The three interfaces and both surfaces of the test specimens were instrumented with No. 30 iron-construction thermocouples. Thermocouple locations are diagrammed in Fig. 8. Strung laterally at the three insulation interfaces and pulled taut by external spring tensioners, the thermocouples, matched for comparative uniformity of reading, were accurate to within ½ F. The radiant lamps were spaced 2½ in. apart with the hot side surface of the test specimen approximately one ft above the lamps. Thermocouple temperature measurements showed that a uniform temperature distribution along the specimen hot-side surface was maintained with this lamp spacing.

Initial tests proved the necessity of adding an "egg-crate" type convection suppressor above the insulation test specimen (see Fig. 9) to prevent transient room air currents from influencing surface temperature readings. Test specimens were made from MIL-B-5924 Type II (with binder burned out), 0.5 lb/cu ft glass fiber and unbonded, 1.5 lb/cu ft, 1000 F temperature limit glass fiber.

EXPERIMENTAL PROGRAM

The first series of tests was conducted with the 0.5-lb/cu ft bonded glass fiber, with the binding agent baked out at high temperature. Steady-state conditions were achieved with one surface of the insulation heated to 600 F and the other subjected to the 80 F room air temperature. Airflow from 0.01 to 0.02 lb/min-sq ft was maintained downward through the insulation. Non-uniform temperature distribution throughout the insulation suggested that strong upward-flowing, free-convection currents were dominant as the mechanism of heat transfer which greatly reduced the benefits expected from the induced downward flow of air. Further study of the 0.5-lb/cu ft insulation was abandoned, and the major portion of the remaining tests involved 1.5-lb/cu ft unbonded glass fiber, having an upper temperature limit of 1000 F.

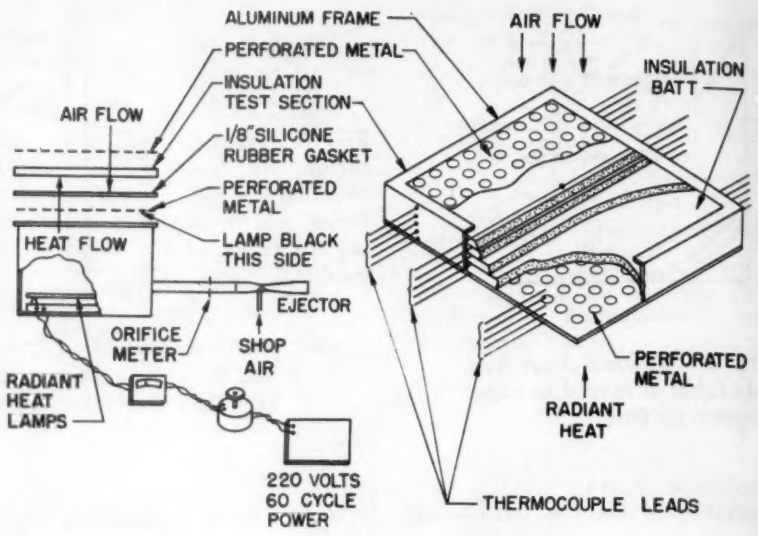
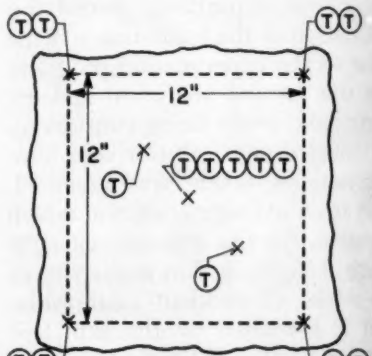


Fig. 7 Test facility and insulation test section

Temperature distribution through the insulation was read at thermal stabilization with the 1.5-lb/cu ft test specimen in a horizontal position, the 600 F hot surface facing down, and with the induced air flowing downward at rates of zero to 0.110 lb/min-sq ft. Pressure drop across the insulation reached 0.15 in. of water at the 0.110-lb/min-sq ft flow rate. A plot of the temperature gradients for various air flows versus distance through the center of the insulation test section is given in Fig. 10.

Shown for comparison purposes is the temperature gradient for zero airflow. Effect of the air in absorbing the inward flow of heat is shown clearly. As airflow is increased, the temperature gradi-

Fig. 8 Center area of the test section, showing thermocouple location



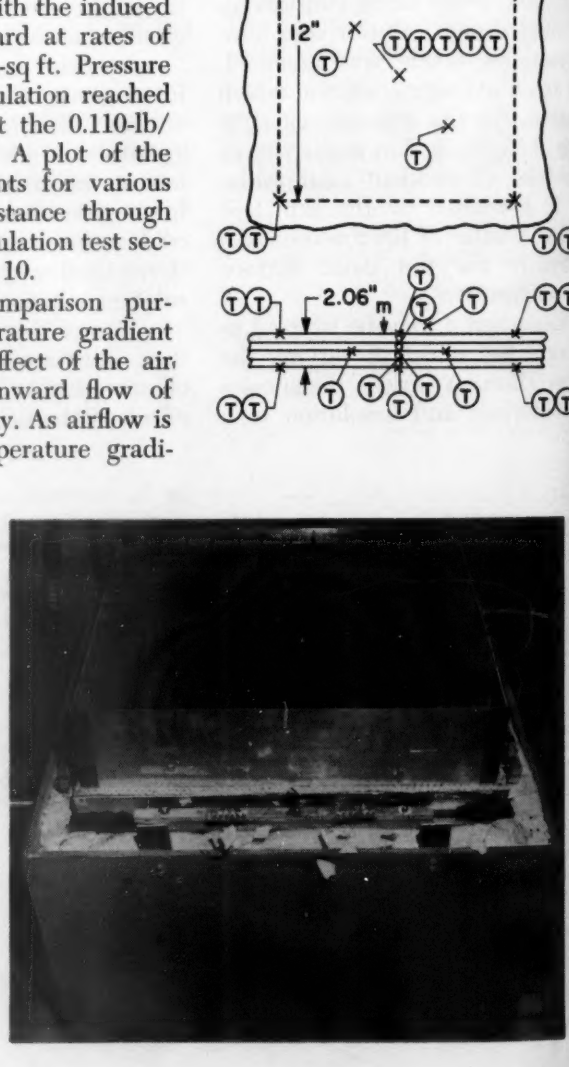


Fig. 9 Convection suppressor added to insulation test specimen

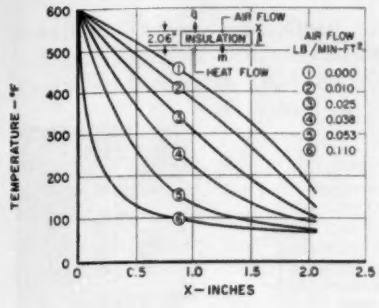


Fig. 10 Temperature gradient through insulation as related to air flow for ambient temperature of 90 F

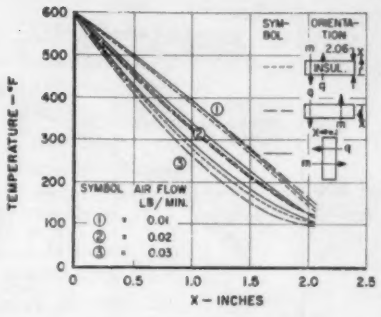


Fig. 11 Temperature gradient through insulation as related to air flow and orientation for an ambient air temperature of 82 F

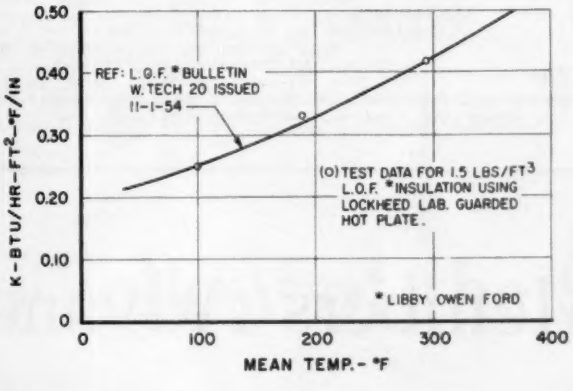

K - BTU/HR - FT² - °F/IN
MEAN TEMP. - °F
REF: L.O.F. *BULLETIN W. TECH 20 ISSUED 11-1-54
(a) TEST DATA FOR 1.5 LBS/FT³ L.O.F. *INSULATION USING LOCKHEED LAB. GUARDED HOT PLATE.
*LIBBY OWEN FORD

Fig. 12 Thermal conductivity of 1.5 lb/ft³ unbonded glass fiber insulation (1000 F limit)

ent is altered so that less heat passes the boundary at x = 2 in., as evidenced by the decrease in the slope of the temperature curve at the cold surface boundary. This means that less heat would enter the aircraft interior as more air is expelled through the insulation. As predicted by the previous analysis, the temperature gradient (and thus the heat flow) at the hot surface boundary, x = 0, increases as the air flow is increased. Since this heat is transferred to the outward flowing air, it does not affect conditions at the boundary, x = 2.

Also investigated was the effect of orientation of the test specimen simulating wall, floor and ceiling installations. With flow rates of approximately 0.01, 0.02 and 0.03 lb/min-sq ft, tests were made to determine the effect upon temperature distribution of orienting the test specimen so that the hot surface was horizontal facing up, horizontal facing down and in a vertical position. Fig. 11 illustrates the comparative results. Resultant temperature gradients were independent of orientation of the test specimen. Therefore, this proposed scheme is practical for insulating bodies such as cylinders, spheres and other typical airborne vehicle shapes.

Test data also were compared with predicted curves calculated from the equations derived previously. Thermal conductivity values required for the analytical solution were determined experimentally with a guarded hot plate accurate to within 3%. Results are shown in Fig. 12, where a comparison is made with the manufacturer's data.

Results of comparison of the analytically predicted performance with experimental data are shown in Fig. 13. Good correlation was achieved above a minimum air flow as the air flow was increased from 0.010 to 0.110 lb/min-sq ft. As stated previously, although the temperature gradient is affected as soon as air flow is induced counter to the direction of heat transfer, the full benefit is not obtained until some minimum flow is achieved. For the insulation under consideration, the minimum flow was estimated to be 0.025 lb/min-sq ft.

Previous testing indicated that glass fiber was, to an extent, translucent to thermal radiation. Although there was some scatter in the experimental data, a value of 0.085 transmissivity was determined. The present test apparatus afforded a rapid, if somewhat crude, method for determining the magnitude of radiant energy transmission through the glass fiber batt when operating at elevated temperatures and its possible effect on the transpiration cooling scheme. As the two-in. batt was comprised of four ½-in. layers, it was convenient to place polished aluminum foil at each interface and to measure its effect on the amount of heat transferred. The heat rate was determined by a conventional calibrated heat meter. Results are presented in Fig. 14. As can be seen, insertion of the radiation shields does not affect greatly the amount of heat transferred through the insulation. For example, assuming a constant heat rate of 40 Btu/hr-ft², the $\Delta T_{1\text{-}5}$ value is approximately 253 F for no reflective shields, 273 F for one layer of foil and 285 F for three layers of foil. The percentage improvement in the apparent thermal conductance is therefore 7.9% for the one-foil configuration and 13% for the three-foil version, confirming roughly the previous data on transmissivity.

SUMMARY AND CONCLUSIONS

The principles of transpiration cooling have been shown to be applicable, with the use of conventional insulating materials, in providing an effective thermal barrier. Heat flowing into occupied compartments and equipment areas may be decreased greatly compared to the heat flow through the same insulation blanket without the transpiration cooling effect.

The main conclusions may be summarized as follows:

1. The simplified analytical model is adequate for predicting the transpiration cooling characteristics of the proposed thermal barrier scheme. Use of these equations enables an optimization study to be made of insulation thickness (weight), exhaust air requirements and refrigeration requirements for desired interior surface temperatures.

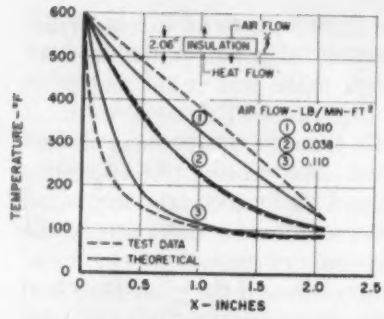


Fig. 13 Correlation of theoretical with test data for an ambient temperature of 90 F

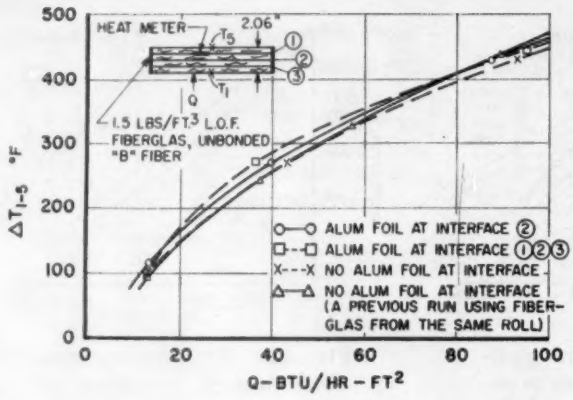

HEAT METER
2.06"
1.5 LBS/FT³ L.O.F. FIBERGLAS, UNBONDED "B" FIBER
ALUM FOIL AT INTERFACE ②
ALUM FOIL AT INTERFACE ①②③
NO ALUM FOIL AT INTERFACE
NO ALUM FOIL AT INTERFACE (A PREVIOUS RUN USING FIBERGLAS FROM THE SAME ROLL)
ΔT1-5 °F
Q - BTU/HR - FT²

Fig. 14 Heat transfer across glass fiber insulation with and without aluminum foil radiation suppression

2. Although theoretically feasible for both open and closed systems, practical considerations limit the application to open or semi-open systems, where either all or a major portion of the transpired air is exhausted from the vehicle.

3. It was determined experimentally that a minimum air flow of 0.025 lb/min/sq ft is required for 1.5 lb/cu ft density conventional glass fiber insulation.

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1. Temperature and Thermal Stresses in Transpiration-Cooled Power-Producing Plates and Tubes, P. V. Schneider, Jet Propulsion, 27, 882, 1957.
2. Regenerative Heat Transfer in Two and Three-Dimensional Flow Through Porous Media, S. W. Churchill, P. H. Abbrecht and Chiao-Min Chu, Industrial Engineering Chemistry 49, 1007, 1957.
3. Transpiration Cooling in Porous Metal Walls, E. Mayer and J. G. Bartas, Jet Propulsion, 24, 366, 1954.
4. A Note on Transpiration Cooling, S. C. Hyman, Jet Propulsion, 26, 780, 1956.
5. Heat Transfer in Sweat-Cooled Porous Metals, S. Weinbaum and H. L. Wheeler, Journal of Applied Physics, 20, 113, 1949.

ASHRAE Members Advanced to

In recognition of unusual distinction in the arts relating to the sciences of heating, refrigeration, air conditioning or ventilation, or to allied arts and sciences, or in the teaching of major courses in said arts and sciences, or of substantial contribution made by reason of invention, research, original work or service as an engineering executive on projects of unusual or important scope, a qualified member in good standing for ten years and over the age of 45 may be honored by elevation to Fellow.

G. L. TUVE

G. L. Tuve, Presidential Member (ASHVE 1948), 16th recipient of the F. Paul Anderson Award and Professor of Mechanical Engineering and Director of Bingham Laboratories, Case Institute of Technology. Since joining the Society in 1932, he has served in many capacities, culminating in the presidency. A three-year term on Council began in 1939 and was followed by his election as Second Vice President in 1946 and First Vice President in 1947. While a member of Council, he served on the Standards, Meetings and Executive Committees, and was Chairman of the latter two. After some years as a member, he became Chairman of the Research Committee and its Executive Committee for 1944 and 1945. Other committee memberships include: Technical Papers, F. Paul Anderson Award, GUIDE Publication, Nominating, Public Relations, Awards and the TACs on Air Distribution and Human Calorimetry. In 1935 he was President of Northern Ohio Chapter, after having served on the Board of Governors.

ALFRED J. OFFNER

Alfred J. Offner, Presidential (ASHVE 1946) and Life Member, and retired consulting engineer. A member of ASHVE Council from 1935 to 1943, he served as Treasurer from 1935-38, was elected Second Vice President in 1944, First Vice President in 1945 and President in 1946. His Committee work includes: GUIDE, Chairman 1922; Admission and Advancement, 1933-35 (Chairman 1935); Regional Meetings, Chairman 1938-39; GUIDE Publication, 1939-42 (Chairman 1942); Finance, Chairman 1939-40; Executive, Member 1941-42, Chairman 1944; Membership, Chairman 1943; Meetings, Chairman 1945; and F. Paul Anderson Award, Chairman 1945. An active member of New York Chapter, he served on the Board of Governors in 1928, 1929 and 1931 and as President in 1930. He was a member of the committees on the New York City Building Code, Building Laws and Regulations Code, Professional Engineering Licensing Law and Heating and Ventilating Exposition. He has written extensively on heating, air conditioning and ventilating.

to Fellow of the Society

SEWELL H. DOWNS

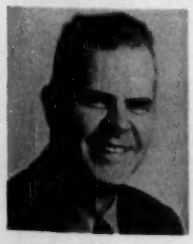
Sewell H. Downs, Presidential Member (ASHVE 1944) and Vice President–Engineering, Clarage Fan Company. Elected to Council in 1936, he advanced to Second Vice President in 1942 and First Vice President in 1943. Committees on which he has served include: Standards (Chairman), Meetings (Chairman), Finance, GUIDE Publication, Research, Ventilation of Garages and Bus Terminals, Air Distribution and Air Friction, Fan Test Code and the Joint Committee on Standards for Residence Heating. Active variously in Western Michigan Chapter, he was Treasurer in 1932-33, Vice President in 1933-34, President in 1934-35 and a member of the Board of Governors from 1935-36. He is the author of several technical articles.

GEORGE V. PARMELEE

George V. Parmelee, Staff Engineer with Arabian American Oil Company in Saudi Arabia, received his education at Fenn College (where he later served as Assistant Professor of Mechanical Engineering), Iowa State and Case Institute of Technology. Prior to assuming his current position, he was Research Supervisor at the ASHRAE Research Laboratory. Other positions in which he has served are as a draftsman for White Motor Company and House Heating Engineer for East Ohio Gas Company. During the period 1945 to 1955 he published approximately 25 research bulletins and papers in ASHVE and ASHAE Transactions, in addition to several papers in other publications. Active locally, he served as Secretary of Northern Ohio Chapter from 1950-52.

WARREN S. HARRIS

Warren S. Harris, Research Professor, Mechanical Engineering, at the University of Illinois, from which he was graduated in 1930 and received his M.S. in 1933. Author of more than 47 technical articles, which have been published variously, he has been most active in research work of the Society. His committee work has included membership on the following: Flow of Fluids, 1943-45; Heating Load, 1945-55 (Chairman, 1951-52); Panel Heating, 1947-57 (Chairman, 1957); GUIDE, 1951-53 (Chairman, 1953); Hot Water and Steam, 1951-59; Infiltration, 1955; Heating and Air Conditioning Loads, 1956-57; and Weather Data, 1958-59. Since the merger, he has served on the Professional Development Committee, the Heating Section of the Divisional Advisory Committee and as a Director-at-Large for the term ending June 1962.



JOHN E. HAINES

John E. Haines, Presidential Member (ASHAE 1955) and Vice President of Minneapolis-Honeywell Regulator Company. Joining the Society in 1940, he was elected to Council in 1950, followed by advancement to Second Vice President in 1953 and First Vice President in 1954. General Chairman of the Committee on Arrangements for the 1949 Semi-annual Meeting, he has served also on the following committees: Chapter Delegates, 1945-46; Nominating, Secretary 1947 and Chairman 1948; Promotion of Research, Chairman 1947-49; Ways and Means, 1951; Membership, Chairman 1952; Chapter Relations, Chairman 1950-51; Finance, Chairman 1953; Long Range Planning, Chairman 1954 and 1959; Executive, Chairman, 1954; F. Paul Anderson Award, Chairman 1954; ASHAE-ASRE Joint Committee on Inter-Society Cooperation; ASHRAE-AIA Cooperation, Chairman 1959-60; and he is currently a member of the UEC Fund Raising Committee, of which he was Vice Chairman in 1959. Active locally, he has been Vice President (1945-46) and President (1946-47) of Minnesota Chapter and is the author of a textbook on controls.

C. O. MACKEY

C. O. Mackey, Professor at Cornell University, began his teaching career in 1924 as an instructor, advancing to Associate and then to full Professor. He has, at various times, been a consultant for the John B. Pierce Foundation, Orange Crush Company, Utica Radiator Corporation and Bendix-Westinghouse Automotive Air Brake Company. Society activities include membership on the Committee on Research, ASHAE Council and the TACs on Cooling Load (Chairman), Glass, Heating and Air Conditioning Loads, Solar Energy Utilization and Panel Heating and Cooling. He is the author of four books: "Graphical Solutions," "Air Conditioning Principles," "Thermodynamic Charts" (with F. O. Ellenwood) and "Engineering Thermodynamics."

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Meetings ahead

August 15-17—Cryogenic Engineering Conference, University of Michigan, Ann Arbor, Mich.

August 21-31 – International Exhibition of New Sources of Energy, Rome, Italy.

August 23-26—International Institute of Refrigeration, Commission 5, Budapest, Hungary.

August 28-September 1—International Heat Transfer Conference, sponsored by the American Society of Mechanical Engineers and the American Institute of Chemical Engineers, Boulder, Colo.

September 20-22—International Institute of Refrigeration, Commissions 2, 3, 6A, 6B and 8, Cambridge, England.

September 27-October 4 – Institution of Heating and Ventilating Engineers, International Heating, Ventilating and Air Conditioning Conference, London, England.

October 2-4 – American Gas Association, Annual Convention, Dallas, Texas.

October 7-10 – Western Building Industries Exposition, Los Angeles, Calif.

October 23-27—National Metal Exposition, Detroit, Mich.

October 31-November 2 – Fourth Canadian Refrigeration and Air Conditioning Show, Toronto, Ont.

November 5-7—National Frozen Food Association, National Convention and Exposition, Bal Harbour, Fla.

November 6-10 – National Warm Air Heating and Air Conditioning Association, 48th Annual Convention, Chicago, Ill.

November 12-15 – Air Conditioning and Refrigeration Institute, Annual Meeting, Hot Springs, Va.

November 13-17—National Electrical Manufacturers Association, Annual Meeting, Atlantic City, N. J.

January 29-February 1—American Society of Heating, Refrigerating and Air Conditioning Engineers, Semiannual Meeting, St. Louis, Mo.

News of ASHRAE members

Honors and Recognitions

J. Donald Kroeker, Fellow ASHRAE, is the 1961 recipient of the F. Paul Anderson Medal, the Society's highest honor, given in recognition of distinguished service. Long active in precedent ASHAE and ASRE, which he joined in 1936 and 1954, respectively, he has served on the following committees: F. Paul Anderson, Nominating, Chapter Relations, Long Range Planning, Standards, Membership, Finance and the TACs on Climate Condition and Heat Pumps. He was a member of ASHAE Council from 1949 to 1951 and Treasurer in 1952. A joint organizer of Oregon Chapter, he was its first President in 1939. Since the merger he has been a member of the Long Range Planning Committee, and is its current Chairman.

Wilmot A. Danielson, retired Brigadier General in the U. S. Army, was awarded the Marston Medal, presented annually to an outstanding alumnus of Iowa State University. A 1907 graduate in electrical engineering, General Danielson served nearly 40 years in the Army, during which he was engaged in construction, maintenance and operation of army installations. He retired in 1946. In addition to membership in ASHRAE, he is a life member of the American Institute of Electrical Engineers and council member of the Association of Applied Solar Energy.

P. B. Gordon, Presidential and Fellow Member, ASHRAE, and Vice President of Wolff & Munier, Inc., has been re-elected Vice President of the Building Research Institute and Vice Chairman of its Executive Committee. He will serve for the term July 1, 1961, to June 30, 1962. Recipient of the F. Paul Anderson Medal for 1960, in recognition of distinguished service to the Society and the industry, he has, in his long career with precedent ASHAE, served in various capacities and on many Society committees.

Bruno P. Morabito, associated with Carrier Corporation, is this year's recipient of the annual Wolverine-ASHRAE Diamond Key Award, presentation of which was made at the 68th Annual Meeting in Denver, held June 26-28. The award was given for his paper, "How Higher Cooling Coil Differentials Effect System Economies," which appeared in the August 1960 issue of the ASHRAE JOURNAL.

New jobs

David M. Lawson has been named Vice President and General Manager for Steveco. District Manager for Sporlan Valve Company for the past thirteen years, he is a member of Atlanta Chapter, ASHRAE, and Field Deputy Director of Refrigeration Service Engineers Society.

Cartis T. Roff, Vice President in charge of production and engineering of Anemostat Corporation of America since 1959, has been elected President. Joining Anemostat in 1946 as Assistant to the President, he was appointed Plant Manager in Hartford, Conn., in 1949 and Vice President in 1959. He is a graduate of Weatherford College.

James A. Hearn, formerly of Way Engineering Company, is going into business for himself as Fox & Hearn, Mechanical Contractors.

John L. Harnack of Southern California Chapter, formerly with United Air Conditioning, has accepted a position with American Air Filter Company. Also of that Chapter, **H. Gordon Paxson** has opened his own manufacturers' representative office as H. Gordon Paxson & Associates and **Hans Sommer** is one of the principals of the newly-formed Western Allied Corporation (he had been Sales Manager of the Commercial Div of Minneapolis-Honeywell Regulator Company).

Albert E. Manning, former International President of Refrigeration Service Engineers Society, has joined Astro Products Corporation as Vice President. Holder of a number of patents and designer of three models of explosion-proof refrigerators, he brings to Astro almost forty years of professional experience. Prior to joining Astro, he was a member of the Occupational Safety Conference of the U. S. Department of Labor. A Charter Member of the Northern New Jersey Section of precedent ASRE, he was also one of the organizers of Garden State Chapter of RSES and the first president of that Chapter.

George C. Briley, formerly National Field Sales Manager of Frick Company, has been named Vice President and Sales Manager of Kerby Saunders, Inc. In this post, he will supervise sales in the fields of air conditioning, heating, refrigeration, plumbing and industrial piping in this country and abroad. Prior to joining Frick, he was southwest regional manager for York Div of Borg-Warner Corporation. He is a graduate of Louisiana Polytechnic Institute.

Richard C. Willson, Jr., appointed Technical Service Manager for Isotron Products, Pennsalt Chemicals Corporation, will be responsible for both the Isotron Engineering Services and Technical Services Laboratories. A 1952 graduate of Johns Hopkins University, he has held various technical service positions with Pennsalt.

William E. Wells is now Field Engineer with the Industrial Div of Dayco Corporation. He joined the company this year after 20 years of experience in design and application of V-belt drives as project engineer for other V-belt manufacturers.

Fred Hermann, with Tenney Engineering, Inc., since 1951, will manage the newly-organized West Coast Div of the company, and becomes Vice President of Harvick Manufacturing Company, where the division will be based. During the past two years he has lectured on thermal and humidity environments at Hofstra College and at Fairleigh-Dickinson, Northeastern and Penn State Universities. He is the author of a number of articles on environmental testing.

John Liebermann, with Ranco, Inc., since 1949, has been appointed Chief Engineer. He was educated at the University of Vienna and the University of London. Also assuming a new post is **Thomas Syfert,** now Product Manager, who has been with the company since 1951. He attended Ohio State University.

E. W. Ervasti, formerly Assistant Director of Marketing for Calumet & Hecla's Wolverine Tube Div, has been appointed Director of Marketing. Beginning his career with the company in 1934, he has worked in various operational and marketing capacities.

Peter Butterfield advanced to the post of Product Sales Manager for packaged chillers, shell-and-tube equipment and fan coil units for Acme Industries, Inc., from his previous appointment as Assistant Field Sales Manager.

Leslie Arnett, named Chairman of Square Manufacturing Company, a subsidiary of Interstate Vending Company, will serve also as Research and Development Consultant. A leading vending machine engineer and designer, he is credited with invention of the first fresh-brewed coffee machine and the first cold drink vending machine. In 1934 he organized the Vendrink Company and in 1946 became one of the organizers of Square Manufacturing Company.

Others are saying—

dust and fume collectors . . . commonly used in the heavy process industries include heavy duty cyclones, fabric filters, scrubbers, electrical precipitators and combinations of these. Problems cited for these units include: plugging of cyclone collectors, either in the vanes or at the dust discharge cone at the bottoms of the collecting tubes; in bag filters, maintenance of shaker systems, bag replacement and blinding of the fabric, causing a rapid increase in pressure drop; build-up of dust and fumes at the wet-dry line junction in scrubbers; and costliness of electrical precipitators, insofar as the original installation is concerned. *Consulting Engineer, May 1961, p 118.*

low temperature technique . . . has become a major tool in the chemical industry, especially in the missile field, liquefied natural gas storage and transportation and helium recovery. As cryogenics has grown, many new techniques and better equipment have become available, such as improved insulation and specially designed valves, pumps, compressors and storage vessels. Cited as the largest area for use of cryogenics in the chemical processing field is production of nitrogen, followed by liquid oxygen and hydrogen. Also mentioned is a process for manufacture of deuterium for heavy water by distillation at −420 F. *Chemical Processing, June 1961, p 28.*

school air conditioning . . . has had much effect on design of educational buildings, such as a tendency to compactness and reduction of window areas. Exploring this subject, this article cites examples of air conditioned schools in various parts of the country, detailing application and building planning. *Architectural Record, July 1961, p. 168.*

PERIODIC HEAT FLOW PROBLEMS CAN BE SIMPLIFIED BY HYDRAULIC ANALOGUES PAGE 45

AT THE DENVER ANNUAL MEETING

(*Continued from page 74*)

William J. Masser acknowledged that product quality is coming to have a high priority among the factors which induce industrial and individual consumers to choose one brand over another. However, technical specifications are becoming more rigorous while quality control costs are rising. It was this problem – demand for quality and rising cost of quality control – which was detailed.

Quality control in operation was the subject of R. D. Sonderup, who covered organization, functions and responsibilities and authorities and their relation to production. Also discussed was the acceptance of the quality control concept by management, by manufacturing, by engineering and design and by marketing groups.

Relationship of cost reduction vs. quality was developed by William E. Mahaffay who traced the historic and paradoxical relationship of cost reduction vs. quality, patterns in engineering education, product development, and marketing during the 1920-1960 era.

At the Forums – There were seven Forums at Denver. R. S. Buchanan was chairman.

These, together with their Moderators, were – Tunnels vs. Burial of Heating, Air Conditioning and Utility Lines, Paul F. Cummings; Introducing and Controlling Humidity in Heated Living Spaces, A. G. Wilson; ASHRAE Research Status, E. P. Palmatier; Industrial Heat Recovery, John H. Clarke; Integration of Lighting and Air Conditioning, W. F. Spiegel; Internal Motor Protectors for Hermetic Refrigerator Compressors, Joseph Kumler; and Problems of High Altitude Installations, D. D. Paxton.

BULLETINS

Temperature Regulators. Extensive engineering information on this manufacturer's line of temperature regulators is contained in eight-page Catalog TCV-1. Available in self or pilot-operated models, the units are illustrated and detailed. Given in the bulletin are features, ratings, dimensions, applications, operation, ranges, materials of construction, sample specifications, flow curve and sizing charts.
OPW-Jordan Corporation, 6013 Wiehe Rd., Cincinnati 13, Ohio.

Ventilators. Included in twelve-page Catalog VC 61-1 are ventilators for wall and ceiling installation and combination ventilator-light and ventilator-light-heater units. Specifications, dimensions and construction details of each ventilator model are included. Color-keyed diagrams illustrate wiring and ducting. Also covered are three models of utility exhaust fans, an automatic shutter and ventilator accessories, such as filters, switches, dampers, timers, wall caps and other fittings.
Fasco Industries, Inc., 255 N. Union St., Rochester 2, N. Y.

Diaphragm Valve Operators. Six-page Bulletin 134-A provides data on eight types of air operators for diaphragm valves. These valves are recommended for remote and automatic flow control of an extensive range of corrosive and non-corrosive liquids and gases. Operators available from stock are listed for valves from ½ through 12 in. Sectional drawings, descriptive information, tables of air actuator pressures for various line pressures, dimensional data and ordering instructions are given for air open-spring close, spring open-air close and air open-air close types of operators.
Hills-McCanna Company, 400 Maple Ave., Carpentersville, Ill.

Sound and Vibration Absorbers. Covering a line of flexible, wire-reinforced rubber sound and vibration absorbers is four-page Catalog 890. Designed and constructed to eliminate noise and vibration disturbance in pipe lines on air conditioning, heating and refrigeration systems, the Sound-Zorber line dampens direct metal-to-metal vibration and, in addition, is cited as absorbing pulsations of the liquid column itself.
General Rubber Corporation, 70 Summit St., Tenafly, N. J.

Relief-Unloading Valve. Described in four-page Bulletin S-5 is a cushioned, automatic relief-unloading valve for steam, air, gas and fluid control service. Sizes 2½ to 24 in. are offered with flanged ends for service to 900 psi. Extensive selection data and engineering specifications are included in the bulletin.
Golden-Anderson Valve Specialty Company, 1232 Ridge Ave., Pittsburgh 33, Pa.

Standby Power Systems. Providing sine-wave ac from storage battery packs, Electro-Pac "A" standby power systems automatically pick up the load when ac line voltage falls below a predetermined level. Describing the units is eight-page Bulletin 6131.
Electro-Seal Corporation, 938 North Ave., Des Plaines, Ill.

Air Pollution Control. Techniques and equipment for air pollution control are presented in 12-page Bulletin 500. New air cleaning precipitators for nuclear submarines, automated controls and high efficiency silicon rectifiers are new products offered. Recent installations are described and research, laboratory, testing and service facilities available are listed and discussed.
Research-Cottrell, Inc., Bound Brook, New Jersey.

Lithium Chemicals. Described in six-page Bulletin 108 are commercial and experimental lithium chemicals for air conditioning and other fields. Properties are listed and both present and suggested applications are given. A selection chart provides a reference guide to important lithium compounds and their industrial uses.
Foote Mineral Company, 18 W. Chelten Ave., Philadelphia 44, Pa.

Application Guides. How to reduce costs and improve existing pressure regulating systems are detailed with diagrams and text in a series of six application guides. Inexpensive ways to provide remote shut-off by a two-position electric motor, remote high-low switchover control, remote pressure control, allocation of loads be-

yond boiler capacity, control of domestic water pressure in tall buildings and load limiting pressure regulation are demonstrated. Diagrams show hook-ups using existing regulators and standard equipment, with specific advantages of each cited.
Spence Engineering Company, Inc., Walden 2, N. Y.

Air Conditioners. Basic specifications of all models in this line are contained in the Fact File; additions to this will be offered periodically, giving specific details on each unit. Comprising the line are four basic series of air conditioning units: MAC, for use under severe operating conditions; SAC, a less expensive version of the MAC Series; TRAC, mobile, two-wheel, trailer-mounted units; and APU, designed for aircraft use, lightweight and engineered for operation on 400-cycle or dc electrical systems.
Airflow Company, 400 S. Stonestreet Ave., Rockville, Md.

Water Chilling Systems. Twenty units with capacities from 50 to 1000 nominal ton are offered for air conditioning and process cooling applications. Physical data and product features are presented in four-page Bulletin L155.10-S.
York Corporation, a Subsidiary of Borg-Warner Corporation, York, Pa.

Electronic Air Cleaner. Electronic precipitation as utilized in the Electro-Cell is a method of removing dust and smoke from the air by use of the attraction between bodies carrying opposite electrical charges. Operation is discussed thoroughly in 16-page Bulletin 258B, which also presents information on construction, maintenance, specifications, dimensions and capacities.
American Air Filter Company, Inc., 215 Central Ave., Louisville 8, Ky.

Pressure Switches. Operation of a line of light-weight pressure and differential pressure switches which actuate at extremely low pressures is described and illustrated in Flyer E7. Data is included also on standard and optional materials, accessories, ports and fluid service. Units in the actuation range from 0.25 to 16 in. of water column operate by means of a spring-biased elastomeric diaphragm on which the differential pressure acts. When the low pressure port is left open to atmosphere, the unit is a pressure switch. Within the pressure range of a given switch, actuation pressure may be altered by turning a direct reading dial.
Pall Corporation, 30 Sea Cliff Ave., Glen Cove, N. Y.

Burners. Presented in Flyer H-10 is the Star-Sixties line of air-tube burners, which are of the air-stream atmospheric type, utilizing air flow to develop high capacity. Units are recommended for air heating applications having maximum ambient air temperatures to 750 F. Detailed in the bulletin are dimensions, specifications, engineering data and ordering information for the four units in the line. Output capacities, examples of burner selection and general application information also are included.
Eclipse Fuel Engineering Company, Combustion Div, Rockford, Ill.

Cooler-Freezer-Refrigerator. Combining a walk-in cooler, walk-in freezer and reach-in refrigerator, Model CF is offered in sizes from six ft ten in. square to 60 ft square. Flyer 553-CF is descriptive of the unit.
Elliott-Williams Company, Inc., 3737 Sutherland Ave., Indianapolis 18, Ind.

STANDARDS PAGE

Heat Pump Standard

A. T. BOGGS, III
ASHRAE Technical Secretary

Revised ARI Standard 240-61 covering Unitary Heat Pump Equipment is now available from ARI at 50c each. The revised standard replaces the 1957 edition and has been rewritten completely to provide a basis for the proposed ARI certification program for unitary heat pumps. The certification program will include unitary equipment up to 135,000 Btu/hr cooling capacity but will not include reverse cycle room air conditioners. Air Conditioning and Refrigeration Institute, 1346 Connecticut Ave., NW, Washington 6, D. C.

A new ARI standard for variable speed transport refrigerated units utilizing forced circulation air coolers has been published by ARI and copies are available from ARI at 75c each. This standard covers equipment such as is used in refrigerated trucks, trailers and railway cars. The new **Standard 1120-61** applies to units intended for high temperature (25 F or above) or low temperature (10 F or below) service. The standard establishes a specification of what constitutes minimum standard equipment; rating requirements; methods of testing; and proper marking for equipment covered by this standard. It does not establish minimum specifications for nor means for determining ratings of refrigerated vehicles.

Department of Commerce: New lists of Commercial Standards and simplified practice recommendations revised to July 1 are available without charge from the Commodity Standards Division, U. S. Dept. of Commerce, Washington 25, D. C. Ask for Catalog #978 List of Commercial Standards, and Catalog #979 List of Simplified Practice Recommendations and Commodity Standards Pamphlet.

Nema: The following revised standards are available from Nema at prices indicated: DC 6-1961 Automatic Temperature Controls (Nameplate Information, Ampere Ratings, Voltage Rating and Load Characteristics), 25c per copy; HE 2-1961 Electric Comfort Heating Equipment, 35c per copy; HU 1-1961 Industrial Heating Units and Devices, 35c per copy. National Electrical Manufacturers Association, 155 East 44th Street, New York City.

Cooling Load: A new form for estimating cooling load requirements for room air conditioner customers has been developed by the room air conditioner section of Nema. This form is used in conjunction with Nema Standard CN 1-1960 covering room air conditioners. The form also includes a map giving climatic differences in all sections of the country which will be of assistance in the proper use of the form. Forms are available in pad form at 50c per pad for 50 copies from Nema Room Air Conditioner Section.

What ASHRAE Regions and Chapters are doing

New officers were installed at meetings held prior to the start of the summer. At technical sessions, speakers discussed classroom air conditioning, short draw copper tubing and air cleaning, among other topics.

LONG ISLAND . . . "Roof Mounted Air Conditioning" was the topic of May speaker John A. Gilbraith of Typhoon.

who's doing what . . . Elected to the Board of Governors were C. Williams and H. Quick.

BRITISH COLUMBIA . . . K. McConnell presented, at the May 10 meeting, a film on short draw copper tubing. Chapter President Fraser Wiggins reported briefly on the Region X Conference, held April 27-28 in Phoenix, followed by a more detailed resume given by Chapter Vice President David B. Leaney.

Main speaker of the evening was Michael Poole, Public Relations Div, Fraser Valley Milk Producers Association, who detailed the formation of the association and its development. His discussion was supplemented by a film.

NIAGARA PENINSULA . . . Operation of the Statronic method of air cleaning was outlined by B. McMichael of C.R.S. Industries at the May 2 meeting. Slides and displays accompanied his presentation.

ILLINOIS-IOWA . . . Problems of putting a man in space were covered by May speaker D. R. Rouser of Bendix Corporation, speaking on "The Human Body Entering Space Flight." Altitude sickness occurs at about 14,000 ft, he stated. With a study of the cryogenic temperature of oxygen, it was found that liquid oxygen can supply man at altitudes of 70,000 ft or higher. Other matters considered were the force of gravity and possible use of algae as an oxygenator.

MEMPHIS . . . Incoming President D. M. Mills introduced these other new Chapter officers at the May 15 meeting: J. P. Hall, 1st Vice President; T. H. Turner, Jr., 2nd Vice President; J. O. O'Brien, Secretary; H. Erb, Treasurer; G. Burr, Historian; G. H. Avery and R. Larkin, members of the Board of Governors.

SOUTHERN CALIFORNIA . . . "Portable Ground Support Air Conditioning, Heating and Ventilating Units in Military Use" was the topic of May speaker James Groff. Illustrating his discussion with slides of different manufacturers' equipment, he covered air-to-air cycle refrigeration units, multipurpose equipment and vapor cycle units. Purposes cited include cooling or heating of electronic equipment and personnel areas. Design suggestions for each of these classes of equipment were offered, with emphasis placed on good distribution of air and refrigerant. Heat of compression, he stated, should not be overlooked where high delivery pressures are involved. Vibration and shock-proof mountings were discussed, and also the use of a pumpdown cycle or crankcase heater to avoid refrigeration migration.

Elected to office were: J. C. Hall, President; B. L. Hutchinson, Jr., Vice President; J. R. Hall, Secretary; V. J. Burke, Treasurer; A. A. Hellmann, R. H. Jorgensen, M. Kodmur and P. A. Van Woerkom, Board of Governors.

FORT WORTH . . . Chapter President G. F. Smith reported at the May 17 meeting on proceedings of the Regional Conference in Shreveport, which was attended also by J. R. Blanke, H. E. Cunningham, A. W. Stubbeman and James L. Tye.

Various aspects of the industry, from the salesman's point of view, were covered by James G. Chatmas, S. T. Lake, R. M. Williamson and James L. Tye.

CENTRAL OKLAHOMA . . . James Wade was speaker at the May 8 meeting.

who's doing what . . . Officers for the 1961-62 season are: W. H. Stewart, President; J. T. McKinney, Vice President; Carl Jensen, Secretary; and M. K. Cohenour, Treasurer.

HOUSTON . . . "Environment for Learning or Climate Control for Classrooms" was the subject of May speaker Maurice J. Wilson, Director of Marketing Development, Carrier Air Conditioning Corporation. Various concepts concerning the air conditioning of schools for year-round use of students were presented. Slides showed a new concept of school construction, cited as lowering the cost of construction and installation of heating and cooling equipment.

Officers for the coming year, as announced at the June 16 meeting, are: President, E. H. McLean (July 1 to December 31, 1961) and J. M. Daniel (January 1 to June 30, 1962); 1st Vice President, J. B. Buckley (July 1 to December 31, 1961) and C. H. Jochen (January 1 to June 30, 1962); 2nd Vice President, J. E. Burton; Secretary, K. C. Gruber; Treasurer, Morris Backer; Board of Governors, E. E. Ransom, Jr., J. T. Ames and W. A. Jackson.

Committee Chairmen are: Membership, J. Ibison and W. Arnold; Program-Public Relations, J. B. Buckley and C. H. Jochen; Arrangements, J. E. Burton; Publications, R. F. Taylor and H. G. Grant;

Attendance and Reception, J. T. Ames and H. E. Conley; Entertainment and Special Events, W. A. Jackson and F. Biggi; Finance, J. B. Hollingsworth; F. E. Giesecke Memorial Fund, H. G. McKee and A. F. Barnes; Houston Engineers' Council, D. D. Price, E. H. McLane, J. M. Daniel and J. B. Buckley; Chapters Regional Committee, C. L. Fleming and H. D. McMillan, Jr.; Student Placement, D. D. Price.

SAN DIEGO . . . Regional Delegate S. A. Bayne reported at the May 9 meeting on the Regional Conference, held in Phoenix April 28-29. Installed as new Chapter officers were: M. S. Jackson, President; W. Neild, Vice President; C. H. Deilgat, Secretary; C. E. Butcher, Treasurer; S. A. Bayne, D. M. Butala and K. Klein, members of the Board of Governors.

NEW ORLEANS . . . Under discussion at the May 16 meeting was "Planning, Financing, Designing, Constructing, Leasing and Operation of a Commercial Building." Guest speaker was Shepard Latter of Latter and Blum, Inc.

ONTARIO . . . Highlight of the June 15 meeting was installation of new officers, who were presented by ASHRAE 1st Vice President John H. Fox. Taking office were: Ernest Fox, President; J. D. Coates, 1st Vice President; and W. A. Mould, 2nd Vice President.

GOLDEN GATE . . . Guest speaker at the June 1 meeting was William Scott, who discussed missile launching pad design and the government space program.

CINCINNATI . . . Installed at the May 16 meeting were: C. P. Wood, Jr., President; C. J. Kummer, Vice President; N. E. Rau, Secretary; J. W. Gibbs, Treasurer; and W. J. Davis, member of the Board of Governors.

EL PASO . . . James Mays of Sporlan Valve Company, guest speaker at the June 19 meeting, utilized slides to supplement a discussion of valves used in heat pumps.

NEW MEXICO . . . Elected to office at the May 16 meeting were V. J. Stephens, President; D. D. Paxton, Vice President; J. L. Desilets, Secretary; and W. L. Beale, Treasurer.

Evaluation of equipment from the standpoints of first cost and performance efficiency was discussed by Max Niese of Public Service Company. A question and answer period followed.

ARKANSAS . . . Speaking at a recent meeting, Stanley Shirley of General Electric Company discussed "Heat Pumps and the Versatility of Application of Heat Pumps." Covered were history and types of installations.

SOUTHERN ALBERTA . . . Outlining the essential parts of a refrigeration cycle, their purpose and operation, Robert Crawford of Canadian Ice Machine Company, Ltd., spoke at the April 18 meeting. A question and answer period followed his presentation.

CHAPTERS REGIONAL COMMITTEE
COMING MEETINGS

REGION I, Central New York Chapter (Syracuse), Oct. 2-3
REGION III, Hampton Roads Chapter (Norfolk), Nov. 4
REGION IV, Jacksonville Chapter (Jacksonville), Oct. 19-20
REGION V, Columbus Chapter (Columbus), Oct. 13-14
REGION VI, Michigan Chapter (Detroit), Oct. 16
REGION VII, Mississippi Chapter (Jackson), Nov. 6

Necrology

J. Donald Preble, who died on May 1 at the age of 60, was Supervisor of Utilities for Northern Illinois Gas Company.

Floyd E. Merryman, Owner of the Mark Agency, died recently in an automobile accident at the age of 49.

Mahlon L. Heist, a Sales Engineer with York Corporation, was 50 at the time of his death.

Frederick M. Jones died at the age of 68. He had been a Member of the Society since 1944.

Henry Norair of National Capital Chapter died on April 12 at the age of 62. He first began his own business in Washington in 1935.

Lloyd F. Hannan, deceased, had been Owner of Ace Sheet Metal Works.

A. Israel, Vice President of Sabal-Kielmann, Inc., was 58 at the time of his recent death.

Thomas J. Abbott, who died recently at the age of 54, had been Vice President of George C. Abbott, Ltd.

Fazal D. Chinoy, deceased, had been Sales Manager for the Bombay Garage Private, Ltd., of Bombay, India.

GREATER PASSENGER COMFORT IN AIRCRAFT IS GOAL OF RECENT STUDIES PAGE 41

ASHRAE ELEVATES SEVEN MEMBERS TO FELLOW OF THE SOCIETY PAGE 88

Candidates for ASHRAE Membership

Following is a list of 98 candidates for membership or advancement in membership grade. Members are requested to assume their full share of responsibility in the acceptance of these candidates for membership by advising the Executive Secretary on or before August 31, 1961 of any whose eligibility for membership is questioned. Unless such objection is made these candidates will be voted by the Board of Directors.

Note: * Advance † Reinstatement

REGION I

New York

BARNES, F. B., Admin. Engr., State of New York, Exec. Dept. Office of General Service, Albany.

D'AVINO, E. A., Engrg. Draftsman III, (Mech.), The Port of N. Y. Authority, New York.

DUNN, J. M., Chief Mech. Engr., Walter Kidde Constructors, New York.

EISNER, HARRY, Vice-Pres., Fiedler Vent & Blower Co., New York.

GIAMBRONE, F. M., Field Engr., Powers Regulator Co., Buffalo.

LEWIS, R. T., Engr., Eastman Kodak Co., Rochester.

POKORNY, L. Z., Research Proj. Engr., General Motors Corp., Delco Div, Rochester.

REGION II

Canada

BARNARD, K. H., Pres., Bon Air Conditioning Co. Ltd., Cornwall, Ont.

BROKENSHIRE, J. M., Mgr. Air Handling Sales, The Trane Co. of Canada, Toronto, Ont.

GARNEAU, ROBERT, Chief Engr., Asselin, Benoit, Boucher, Ducharme, Lapoint, Montreal, Que.

KROVATS, PAUL, Project Mgr., Smith, Carter, Searls Assoc., Winnipeg, Man.

MONK, A. C., Mech. Repr., National Heating Prod., Rexdale, Ont.

PARRATT, J. A., Plant Engr., International Business Machines Co., Ltd., Toronto, Ont.

WARD, G. T.,* Dir. of Research, Brace Bequest, Faculty Engrg., McGill University, Montreal, Que.

WREFORD, H. W., Sales Repr., The Trane Co. of Canada, Montreal, Que.

REGION III

District of Columbia

FRIEDLAND, MARCEL,* Gen. Engr., Federal Power Commission, Washington.

Maryland

FOARD, C. H., Design Engr., Miller, Schuerholz & Gipe, Baltimore.

Pennsylvania

DEMAND, E. E., Project Engr., Philco Corp., Philadelphia.

MAGUIRE, J. M., Vice-Pres., Penn Warner, Inc., Chambersburg.

MITCH, C. J., Engr., Buchart-Horn, York.

WALTON, J. E., SR., Sales Engr., R. D. Bitzer Co., Inc., Melrose Park, Philadelphia.

Virginia

HAYES, W. C.,* Estimator & Engr., Horne Bro. Inc., Newport News.

REGION VI

Georgia

LEWIS, O. K., JR., Mech. & Elec. Cons. Engr., Albany.

ROESEL, A. J., Pres., H. A. Sack & Co., Statesboro.

North Carolina

GAGGART, H. M., JR.,* Design Engr., Davis Mechanical Contractors, Inc., Charlotte.

REGION V

Indiana

GALBREATH, J. W.,* Engr., Walker, Applegate, Oakes & Ritz, New Albany.

HOTELL, C. D., Mgr. Refr. & Htg., F. H. Langsenkamp Co., Indianapolis.

NORMAN, K. L., Engrg. Indus. Spec., The Trane Co., Indianapolis.

RICHARDS, G. W., Supvsr. Customer Service, Whirlpool Corp., Evansville.

Ohio

ANDREW, G. W., Vice-Pres., Engrg., Keco Industries, Inc., Cincinnati.

ARMSTRONG, J. R., Project Engr., Standard Plumbing & Heating Co., Canton.

BLACK, R. L.,† Br. Mgr., Johnson Service Co., Akron.

BUCKLEY, E. J., Partner, Scheeser & Buckley, Akron.

CULLEY, R. M., Mech. Engr., Goodyear Aircraft Corp., Akron.

CUNNINGHAM, C. D., Owner, Akron Ventilating Co., Akron.

FERGUSON, J. H.,† Sales Engr., Johnson Service Co., Akron.

GEOPFERT, T. J., SR., Engr., J. W. Geopfert Co., Akron.

GRIGGS, R. L., Vice-Pres., H. & C. Supply Co., Inc., Akron.

HALLEY, R. T., Br. Mgr., Allied Equipment Co., Cuyahoga Falls.

HOERBELT, G. J., Technician, East Ohio Gas Co., Akron.

HOTTENSMITH, A. C., Estimator, Kraws P. & H. Co., Akron.

IRELAND, N. J.,† Design Engr., B. F. Goodrich, Akron.

JENKS, N. R., Refr. Supvsr., Spohn Heating & Vent., Akron.

JONES, J. O., Chief Engr. Design & Sales, Middleton Plbg. Co., Akron.

KERR, R. J., Vice-Pres. & Treas., Temperature Control Co., Akron.

KOCHEL, R. K.,* Htg. & Vtg. Group Leader, Goodyear Aircraft, Akron.

LOTZ, W. A.,* Testing & Tech. Service Engr., Owens-Corning Fiberglas Corp., Granville.

MARKELL, M. C., Engr., Spohn Heating & Vent., Akron.

OLDFIELD, R. L., Sales Engr., Minneapolis-Honeywell Reg. Co., Akron.

PALMER, S. E., Estimating & Supervision, Engrg. Dept., Air Comfort Inc., Akron.

PETTY, J. G.,* Chief Engr., Davey Sales Co., Akron.

PFAHL, C. A., JR.,* Vice-Pres. & Mgr., XXth Century Furnace, Inc., Akron.

PIDWELL, J. C., President, The H. P. Cahill Plumbing Co., Akron.

REDMOND, R. C., Sales Engr., Sherman & Weager, Cleveland.

RICHARD, T. C., Plt. Engr., Sr., Hgt. & Vtg., Goodyear Aircraft, Akron.

ROSS, P. W., Technician, East Ohio Gas Co., Akron.

SMITH, J. P., Vice-Pres., Refrigeration Supplies, Inc., Akron.

SIEGFERTH, R. H., Owner & Pres., Siegferth Heating & Plumbing Co., Inc., Akron.

VIALL, J. R., Owner, Akron Plumbing & Heating Co., Akron.

WILLIS, D. R., Vice-Pres. & Treas., Dan H. Willis Co., Akron.

REGION VI

Illinois

ATKINSON, J. J., Sales Engr., Powers Regulator Co., Chicago.

BIRMINGHAM, R. J., Design Engr., General Electric Co., Chicago Heights.

WALTNER, M. D., JR., Chicago Br. Mgr., Preferred Utilities Mfg., Chicago.

South Dakota

GREENSLATE, H. G., Chief Engr., Fenn Bros., Sioux Falls.

GROTE, D. R., Partner, Milbrandt & Grote, Aberdeen.

MILBRANDT, K. G., Partner, Milbrandt & Grote, Aberdeen.

REGION VII

Louisiana

POL, W. I.,* Vice-Pres. & Gen. Mgr., Airtrol Engineering Co., Baton Rouge.

Missouri

SCHMIDT, W. A., Sales Engr., Worthington Corp., St. Louis.

REGION VIII

Oklahoma

NOLTE, E. R., Sales Engr., Gordon's Specialty, Norman.

Texas

BLANKE, J. R.,* Vice-Pres., Blanke Mechanical Contrs., Fort Worth.
BLUM, H. A., Professor, Engineering School, Southern Methodist University, Dallas.
MCLANE, T. C., Engr., Frostmaster Co., Tyler.

REGION IX

Colorado

DIEHL, O. E., Chief Engr., Clarke-Pacific, Inc., Denver.

Nebraska

BLUE, A. D., Cons. Engr., Omaha.
MCELROY, J. L., Vice-Pres., The McElroy Co., Inc., Grand Island.
THOMAS, R. C., Mech. Equip. Designer, Davis & Wilson, Lincoln.

Texas

JACKSON, G. H.,* Partner, Herlin Engineering Co., El Paso.

REGION X

California

SCOFIELD, C. M., Sales Engr., American Standard, Ind. Div, San Francisco.

Washington

BOULANGER, L. F., Facilities Engr., Boeing Co., Seattle.

FOREIGN

Africa

GARDNER, H. W., Refr. Mgr., G. B. Ollivant, Ltd., Accra, Ghana.
GILLHAM, A. E., Sr. Sales Engr., United Africa Co. (Tech.) Ltd., Lagos, Nigeria.
HASLAM, L. A. K., Area Engr., Haden Nigeria Ltd., Yaba, Lagos, Nigeria.

Australia

HOPKINS, R. E., Engr., W. E. Bassett & Partners, Melbourne, Victoria.

Brazil

PESSOA, P. A., Frigidaire Mgr., General Motors do Brasil, Sao Paulo.

Chile

SELL, C. W., Engr., Soclima S. Carroasco S.A.C., Santiago.

England

CLARK, KENNETH, Chief Engr., J. G. L. Poolson, Pontefract, Yorkshire.

POOLE, HAROLD, Mgr. A-C Div, Brightside Heating & Engineering, Ltd., London.

TOWNSEND, LAWRENCE, Design Engr., Shell-Mex & B. P. Ltd., London.

France

MOREAU, MAX, Chief Engr., E. Moreau, Pau.

India

JAIN, N. L.,* Reader in Mech. Engg. Institution Govt. Engg. College Jabalpur.

Italy

ANGELI, FULVIO, Vice-Pres., Dell'Orto Chieregatti Co., Milan.
KLAT, ELIE, Cons. Engr., Milan.
MARABELLI, ANGELO, Air Cond. Sec. Dir., Dell'Orto Chieregatti Co., Milan.
PONTI, GIUSEPPE, Tech. & Com. Mgr., Officine Di Seveso S.p.A., Milan.
SHEET, H. A., Engr., Government of Iraq, Rome.

Sweden

ABJORNSSON, L. N., Sales Mgr., Ab Bahco, Ventilating Div, Enkoping,

STUDENTS

CRELLIN, G. L., Dartmouth College, Hanover, N. H.
GUTOW, B. S., University of Illinois, Urbana, Ill.
NELSON, R. C., Auburn University, Auburn, Ala.

BULLETINS

Electric Unit Heaters. Information on heaters for office, store and industrial use is provided by four-page Bulletin DB5-100. Specifications and illustrations are included and controls, both manual and automatic, are introduced.
Ilg Electric Ventilating Company, 2850 N. Pulaski Rd., Chicago 41, Ill.

Electrostatic Precipitators. Discussing the air pollution problem and application to it of these units, twelve-page Bulletin 2128 describes principles of operation. General specifications for the line are outlined and illustrated with cutaway drawings. Also covered are electronic controls and a high voltage rectifier.
American-Standard, Industrial Div, Detroit 32, Mich.

Noise Control. Approaches to solving acoustical problems in appliances and equipment are outlined in 14-page Bulletin 1-EA-910-A, "Appliance and Equipment Quieting Materials." Both absorption of air-borne noise and reduction of structurally-borne sound are explained. The information is documented with details of test methods used to arrive at basic data presented. Performance tables, nomographs, drawings and photographs supplement the text.
Owens-Corning Fiberglas Corporation, National Bank Building, Toledo 1, Ohio.

Oil Burner Primary Control. So compact it can be mounted on a four by four-in. junction box, eliminating any need for special enclosures, the new Type 668 is the subject of four-page Bulletin R-1700. Instantaneous response to both flame ignition or extinction is provided by the unit, which is not affected by the radiant heat of a red-hot combustion chamber or the ignition spark.
White-Rodgers Company, 9797 Reavis Rd., St. Louis 23, Mo.

Motor Operated Valves. Quiet operation, compact size, easy installation, high flow capacities, simple design, 100% shut-off and fail-safe operation are cited for these two and three-way, motor-operated valves for fan coil units. Included in six-page Bulletin 210 are product features, flow diagrams, operating and construction specifications, flow versus pressure drop curves and dimensional diagrams.
Erie Manufacturing Company, 4000 S. 13th St., Milwaukee 21, Wisc.

Vacuum Heating Pumps. Comprehensive performance and selection tables for the VCA line of vacuum pumps are presented by four-page Bulletin S153. Sizing information is included and product features are detailed.
Domestic Pump & Manufacturing Corporation, Shippensburg, Pa.

Indicators, Controllers. Operating principles, two or three-position control features, applications, small space installation, servicing and special design features of two models (potentiometer or bridge) of indicators and indicating controllers are provided in four-page Bulletin 67. Further information is offered on components: the High-Gain Servo-Amplifier and motor, reference junction compensator, constant voltage supply, slidewire assembly and the more than fifty ranges and calibrations available. Included also are specifications on sensitivity, accuracy, response speed and operating requirements.
Thermo Electric Company, Inc., Saddle Brook, N. J.

Northern Alberta Chapter Achieves its UEC Quota ...FIFTH ASHRAE CHAPTER TO MEET ITS GOAL and First From Canada

Northern Alberta has raised its $900 UEC Fund quota and, by so doing, is the fifth ASHRAE chapter to meet its goal and the FIRST from Canada and Region II.

On June 16, R. A. Williams, ASHRAE-UEC Building Fund Chairman for the 60-member Northern Alberta Chapter, wrote:

"It is with a great deal of pleasure that I enclose a cheque for the sum of $900, on behalf of the Northern Alberta Chapter, for our assigned quota of the UEC Building Fund.

"We hope that other Canadian chapters will join us at arriving at their quotas in the near future."

Of the total contribution, $825 has been given individually by members and the remainder came out of the Chapter treasury.

The other successful chapters to achieve their UEC quotas were Central Oklahoma — Region VIII (ASHRAE JOURNAL, June 1960, Page 59); South Piedmont — Region IV (ASHRAE JOURNAL, October 1960, Page 80); Toledo — Region V (ASHRAE JOURNAL, March 1961, Page 78); and Tucson — Region X (ASHRAE JOURNAL, July 1961, Page 89).

As of July 1, ASHRAE had raised $46,529.84 towards our pledged quota of $250,000. We are making substantial progress, but if we are to meet our goal we have a great deal of work to do. There are still ten chapters who have not yet organized Fund Raising Campaigns: Region II — Niagara Peninsula and Southern Alberta; Region III — Baltimore, Central Pennsylvania and Hampton-Roads; Region VI — Wisconsin and Central Michigan; Region VII — St. Louis; Region VIII — Alamo and Fort Worth.

There are many new and beautiful skyscrapers in New York City. All of them required bricks, mortar and steel in their construction. Notable among them, the United Engineering Center . . . which will be ready for occupancy late in August . . . also includes the ingredient PRIDE within its walls, represented by the contributions of thousands of engineers throughout the United States and Canada, as well as from other countries.

HAVE YOU CONTRIBUTED YOUR SHARE?

One Man's Opinion

The following statements were contained in a recent letter from Associate Member William L. Osborn, who is now residing in Waynesboro, Pa. They are repeated here in order to give JOURNAL readers one man's opinion regarding voluntary and individual financial support to the United Engineering Center building fund. Mr. Osborn, who is also a member of ASME, contributed generously to this fund.

I have been a member of both ASME and ASHRAE (including its parent societies) for nine years. Presently, I am located more than sixty miles from the local chapters of either society and HAVE NEVER BEEN PERSONALLY CONTACTED WITH REGARD TO THE UEC FUND.

However, I feel compelled to make this contribution because, when we who raise the plaintive cry—"but, I'm a professional"—do not support something as fundamental as this, when we default on this most demonstrative proof of membership in a profession—then it can only be that we fail to understand what it is that we so ardently proclaim. Every preceptial arch of this or any profession is held aloft by the keystone of individual self-sacrificial advancement of that profession by its members. If we are to be differentiated from 'train drivers', it must be by emulating those who have already given of themselves so generously in the cause of the United Engineering Center Building Fund.

With these few dollars and many words, go my heartfelt and sincere wishes for the complete success of this undertaking.

The new United Engineering Center is rapidly nearing completion and the staffs of 19 engineering societies, including ASHRAE, are looking forward to relocating in this building by September.

HAVE YOU DONE YOUR SHARE IN BUILDING THIS ENGINEERING SYMBOL?

NAME .....

ADDRESS .....

IN CONSIDERATION OF THE GIFTS OF OTHERS INTENDS TO GIVE TO

UNITED ENGINEERING CENTER BUILDING FUND

.....DOLLARS $.....

BALANCE TO BE PAID QUARTERLY $....., SEMIANNUALLY $.....,

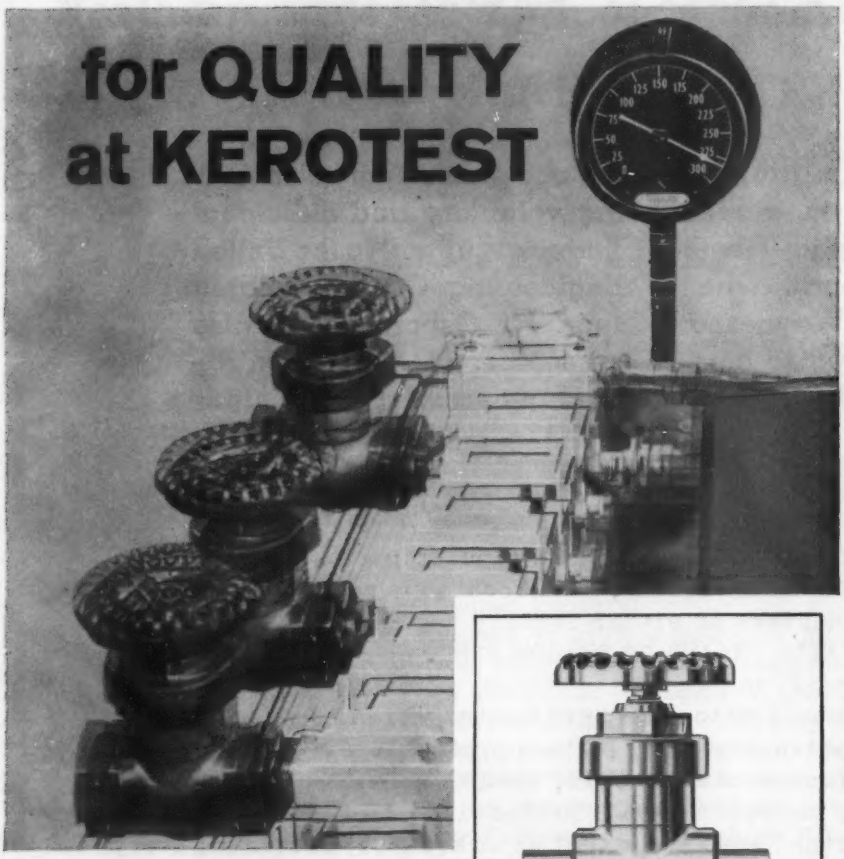
ANNUALLY $....., OR AS FOLLOWS .....

SIGNED.....

MEMBER ASHRAE

CHECK MAY BE MADE PAYABLE TO UNITED ENGINEERING TRUSTEES, INC.
29 WEST 39TH STREET, NEW YORK 18, N. Y.
GIFTS ARE DEDUCTIBLE FOR INCOME TAX PURPOSES

SHAKEDOWN



Vibration-proof Construction

Only precision, quality-controlled construction can take this kind of concentrated abuse. These valves, under a pressurized refrigerant, are vibrated at 3600 cycles per minute for prolonged periods. At intervals throughout the accelerated test, Kerotest engineers evaluate performance and condition of seat seals, diaphragm seals, solder seals and handwheels.

The high standards of material quality and craftsmanship maintained at Kerotest assure you of smoother valve operation and minimum maintenance. Tested and proved design features, quality control at each step of manufacture combine to make Kerotest the best valve that money can buy.

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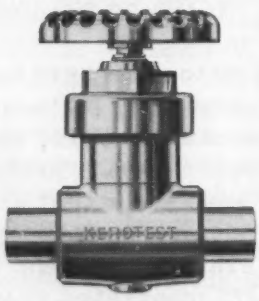
ASK FOR KEROTEST VALVES AT YOUR LOCAL WHOLESALERS

KEROTEST MANUFACTURING CO.

2510 Liberty Ave. • Pittsburgh 22, Pa.



KEROTEST 520 DIAPHRAGM PACKLESS GLOBE VALVE

The precision-built brass valve for all types of refrigeration and air conditioning systems. Ideal for liquefied petroleum gas bulk plants, pipe line loading racks, laboratory installations, nitrogen, oxygen (degreased) and compressed air applications.

Specify the Kerotest 520 globe valve for operating pressures up to 400 psi and temperatures up to 200° F. Use it in vacuum service applications to 1 micron.

KEROTEST QUALITY CONTROLLED FEATURES

- Replaceable spindle seats for continuous positive seating
- Metal to metal backseating above and below diaphragms
- Beryllium copper and stainless steel diaphragms
- Generous handwheel

KEROTEST

BULLETINS

Filter-Driers. Describing and illustrating how Molecular Sieve Filter-Driers of eight different types provide drying, filtering and acid removal in refrigeration and air conditioning systems, six-page Bulletin 276-A is now available. Principle features of the units are outlined, supplemented by photographs and cross-sectional views, and specification tables are presented for each type. Listed in a performance data table are connection sizes, pressure drop, psi and drying capacities.
American-Standard, Controls Div, 5900 Trumbull Ave., Detroit 8, Mich.

Valve Line. Cutaway drawings, materials of construction charts, pressure-temperature curves and dimension tables are included in a 16-page catalog descriptive of the industrial Flo-Ball valve line. Featured are several series of top entry, 150-psi flanged valves; 300-psi flanged valves; 600-psi WOG screwed end, bar stock valves; and vacuum to 3000-psi shut-off valves.
Hydromatics, Inc., 5 Lawrence St., Bloomfield, N. J.

Unit Cooler. Various models in the Lo-Aire line are illustrated and discussed in eight-page Bulletin 260. For ceiling mounting, these units are supplied with hangers attached to the casing, which automatically provide the pitch required to drain the condensate toward the drain connection. Use of a double drain pan is cited as eliminating drain pan sweating problems.
Bohn Aluminum & Brass Corporation, Danville Div, Danville, Ill.

Air Curtain. To prevent loss in industrial plants having large door openings, a curtain of hot air across the opening can be provided by a direct gas-fired, air heating system. Typical installations are pictured in Flyer D-C-1.
Maxon Premix Burner Company, Inc., Muncie, Ind.

Coupling Application. Eight-page Bulletin WB-1, "How, When and Why to Use a V-Band Coupling," explains how the simplified fastening of this type of coupling can improve functional design and product appearance, decrease assembly time and reduce manufacturing costs. Available coupling and flange configura-

(*Continued on page* 103)

Contact your nearest agent
for further information about 'Arcton'
chlorofluorohydrocarbons.

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I.C.I. of A. & N. Z. Ltd., P.O. Box 1911, Melbourne, C.2.

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I.C.I. (Belgium) S.A., 32 Rue Edmond Tollenaere, Brussels, 2.

CHILE
Cia Imperial de Industrias Quimicas de Chile S.A. Com. e Ind., Casilla 1357, Santiago.

COLOMBIA
John Simon & Cia. Limitada, Calle 14 No. 12-50, Oficina 815, Bogota.
Rodrigo Agudelo O, Edif. Garces, Avenida Colombia Calle 11, Oficina No. 303, Cali.
Walter Bridge & Co. Ltd., Edificio "Jenaro Gutierrez", Calle Colombia No. 47-28, Medellin.
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Holger Andreasen, A/B, Storgatan 7, Orebro.

URUGUAY
Industrias Quimicas Uruguayas "Duperial", Avda. General Rondeau 2050, Montevideo.

VENEZUELA
H. Kern & Co., S.A., Apartado 1567, Edificio Kern, Av. Ppal. Los Cortijos de Lourdes, Caracas.

ICI GENERAL CHEMICALS DIVISION

BULLETINS

(*Continued from page* 100)

tions are illustrated and applications are detailed.
Aeroquip Corporation, Marman Div, 11214 Exposition Blvd., Los Angeles 64, Calif.

Boilers. Describing Kewanee Square-Heat Type R Series boilers for heating medium-size buildings, is eight-page Bulletin 549B. Included are tabulated data on ratings and dimensions of the units, which range in capacity from 243,000 to 1,350,000 Btu/hr. Also covered are features, typical specifications and extensive selection data for available heating coils and induced draft fans.
American-Standard, Industrial Div, Detroit 32, Mich.

Lubricated Plug Valves. Calculation of pressure drop through lubricated plug valves is outlined by twelve-page Bulletin 198, which compiles data gathered on existing valves. Listed are 85 valve figure numbers from which the engineer may select data and calculate pressure drop with either the K-values or equivalent feet method.
Walworth Company, 750 Third Ave., New York 17, N. Y.

Methyl Chloride. Storage, transfer and measurement of methyl chloride are covered in a 16-page bulletin, cited as applying equally well to other liquefied gases. Among the subjects dealt with are handling and supporting cylinders, emergency handling of leaking containers, pressure in a container, transfer by heat and transfer by compressor.
Ansul Chemical Company, Chemical Products Div, Marinette, Wisc.

Roof-Top Air Conditioner. Completely packaged, fully insulated, charged and factory-tested, these roof-top, year-round air conditioning units are ready for installation. By use of a multiple system, each unit is installed directly above the area to be conditioned, eliminating ducts. Descriptive of the line is 24-page Bulletin RT-B1.
Melchior, Armstrong, Dessau, Inc., Ridgefield, N. J.

Valve Line. Featured in 24-page Catalog MV-615 are 28 categories of valves for industrial, commercial and domestic applications. Details, specifications, diagrams and roughing-in dimensions are provided for each valve, 74 of which are illustrated. Among items included are: angle valves for copper tubing, iron pipe, radiator and heavy duty; and convector valves.
Marsh Valve Company, 104 Brigham St., Dunkirk, N. Y.

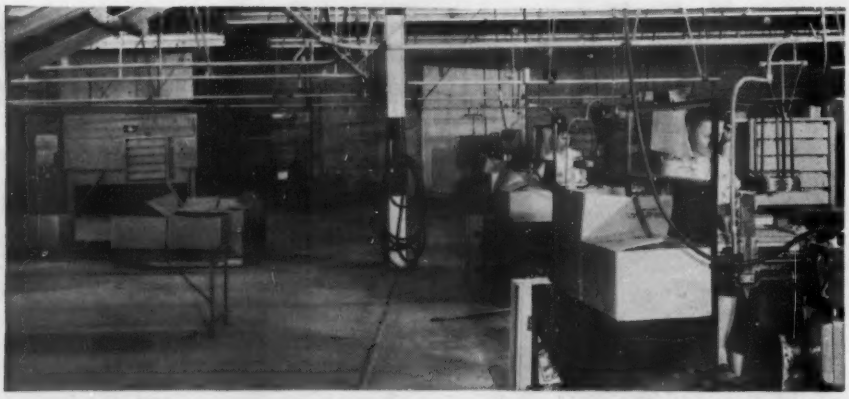
Flexible Coupling. Three types of a new flexing member coupling, together with the Type DE Gearflex coupling are offered by this manufacturer. Stocked units of the couplings are available for shaft diam from ½ to 3½ in. and for hp/100 rpm requirements from 0.3 to 29.0. Descriptive of the line is a twelve page catalog.
Lovell Manufacturing Company, Flexible Coupling Div, Erie, Pa.

Power Roof Ventilators. Superseding Bulletin 2303, eight-page DB3-100 covers this company's line of propeller fan, power roof ventilators. In addition to a new size, more recent information, new illustrations and revised dimensions have been incorporated in the catalog.
Ilg Electric Ventilating Company, 2850 N. Pulaski Rd., Chicago 41, Ill.

Zone Control Valves. Installation of motorized Zone-A-Trol Straight Flo Valves to provide individual room temperature control or zone control on any new or existing hydronic heating system is discussed in a four-page bulletin. Provided are a number of piping layout diagrams for zoning various types of systems and step-by-step installation is illustrated.
Viking Instruments, Inc., Econo Products Company Div, East Haddam, Conn.

Split-Body Control Valves. Details on construction and operation of the single-seated, split-body LB valve are provided by 16-page Bulletin LB-4. Optional actuators and accessories are shown, features are described and extensive specifications and valve sizing formulas are given.
Conoflow Corporation, 2100 Arch St., Philadelphia 3, Pa.

Air Valves. Three new air valve models capable of filling and exhausting a 100-cu in. load chamber from zero to 90 psi in 0.150 sec are illustrated and described in four-page Bulletin 0611-B1. Each valve is discussed briefly in relation to its uses in varied industrial applications. In addition, a companion line of bottom and side-ported bases is covered.
Hannifin Company, 501 S. Wolf Rd., Des Plaines, Ill.



DRY and CLEAN AIR at the RIGHT TEMPERATURE

For Processes or Material Demanding Precise Control, Use Niagara Liquid Absorbent Air Conditioning.

This compact method, giving high capacity in small space, removes moisture from air by contact with a liquid in a small spray chamber. The liquid spray contact temperature and the absorbent concentration, factors that are easily and positively controlled, determine exactly the amount of moisture remaining in the air.

Most effective because . . . it removes moisture as a separate function from cooling or heating and so gives a precise result, and always. Niagara machines using liquid contact means of drying air have given over 20 years of service. The apparatus is simple, parts are accessible, controls are trustworthy.

Most reliable because. . . the absorbent is continuously reconcentrated automatically. No moisture-sensitive instruments are required to control your conditions...no solids, salts or solutions of solids are used and there are no corrosive or reactive substances.

Most flexible because...you can obtain any condition at will and hold it as long as you wish in either continuous production, testing or storage.

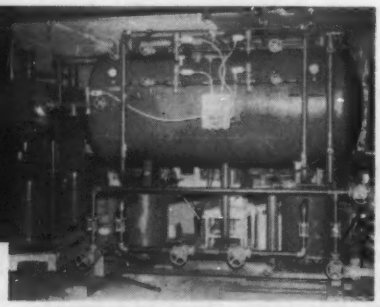
Write for Bulletins 112 and 131 and complete information on your air conditioning problem.

NIAGARA BLOWER COMPANY

Dept. RE-8 405 Lexington Ave., New York 17, N. Y.

Niagara District Engineers in Principal Cities of U.S. and Canada

THE ULTIMATE IN THE COMPRESSION REFRIGERATION CYCLE*



THIS IS ANOTHER CYCLE CENTER, factory assembled and on its way to a 400 ton milk and ice cream plant.

This 400 ton CYCLE CENTER also eliminates the intercooler required in two-stage systems.

WHAT WILL IT DO?*

It will provide liquid overfeed to the evaporators, catch the excess liquid and recirculate it to the evaporators, with these results:

- FULL COMPRESSOR PROTECTION AGAINST SLUGS
- PEAK COIL AND COMPRESSOR EFFICIENCIES
- SUB COOLED LIQUID FEED AT CONSTANT PRESSURE THE YEAR AROUND
- PRACTICALLY UNLIMITED RATE OF LIQUID FEED AT ABSOLUTELY NO POWER COST
- NO MECHANICAL PUMPS
- NO FLASH GAS IN LIQUID LINES
- SAFE, AUTOMATIC PLANT OPERATION
- OIL SEPARATION, ANY REFRIGERANT
- HIGHER SUCTION PRESSURES
- LARGE POWER SAVINGS

A similar 400 ton CYCLE CENTER after installation and insulation.

- LARGE SAVINGS IN FIRST COST ON NEW PLANTS. FOR EXAMPLE, THE RECEIVER IS NOT REQUIRED AND SURGE DRUMS ARE ELIMINATED.
- AUTOMATIC HOT GAS DEFROSTING AT MINIMUM COST.

ASK FOR BULLETIN CC-2

***NOT JUST A LIQUID RETURN UNIT.**

Available for any refrigerant, in capacity from 10 to 1,000 tons and more. Factory assembly is optional.

J. E. Watkins Co.

1311 SOUTH FOURTH AVENUE
MAYWOOD, ILLINOIS

NEW PRODUCTS

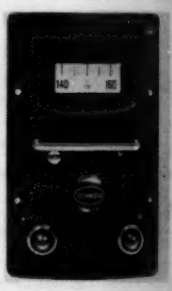
DOMESTIC BOILERS

Two new sizes have been added to this line of domestic boilers; available for firing by oil or gas, they have inputs of 5.5 and 9 gph or 770,000 and 1,260,000 Btu/hr. Models L-36 and L-48 feature large boiler water storage area which, combined with large size copper instantaneous tankless coils, makes them suitable for applications where a continuous flow of hot water is required.

Columbia Boiler Company of Pottstown, Pottstown, Pa.

TEMPERATURE CONTROLLER

Of the resistance bulb signalling type, the Pt Temperature Controller offers a choice of on-off or proportioning control over the range from −150 to 1050 F. Interchangeable scales are printed on stainless steel in one-deg graduations and 300-deg spans, for both F and C.



One probe handles all ranges. Other features include transistorized circuitry, platinum resistance thermometer sensing element, plug-in construction and low power requirements.

Duncan Electric Company, Inc., Technique Associates Div, 1413 N. Cornell, Indianapolis 2, Ind.

BLOWER HOUSING

For heating and cooling systems, this blower housing is constructed entirely of glass fiber and is molded to the exact shape of the unit on which it will be used. A vinyl lining on the inside adds strength and at the same time serves as a vapor barrier. Noise from the blower is reduced by the glass fiber, which also is cited as assuring fire resistance.

Fibrous Glass Products, Inc., a Subsidiary of Pall Corporation, Alpa Plaza, Hicksville, N. Y.

INDUCTION CONDITIONERS

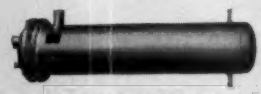
Featured on this new line of induction room air conditioners is a front panel that is unbroken by recirculated air grille work. Secondary air is drawn in through a concealed opening in the kick space under the unit. Outlet grilles are concealed in the cabinet

(*Continued on page* 108)

M-PAK MODULAR PACKAGES... A NEW ADDITION TO B&G® AIR CONDITIONING AND REFRIGERATION EQUIPMENT

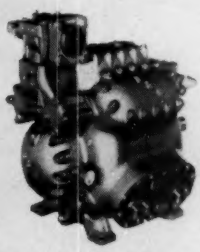
A COMPLETE LINE OF REFRIGERATION AND AIR CONDITIONING COMPONENTS

B&G CONDENSERS
Lighter, more compact units, featuring quiet, vibration-free operation, uniform gas distribution, cleanable tubes and higher operating efficiencies.

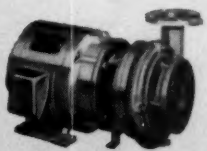


B&G EVAPORATORS
Featuring non-slugging design...tube bundle proof against freeze-up damage ...positive oil return.

B&G SUCTION LINE HEAT EXCHANGERS
Aid thermal expansion valve operation, prevent liquid slugging.



B&G COMPRESSORS
Precisely engineered for vibration-free operation with all the advantages of "open" units. All units have *one* bore —a few interchangeable parts service them all.



B&G CENTRIFUGAL PUMPS
A full range of *quiet* pumps for refrigeration and air conditioning systems.

Modular packages increase availability...speed-up delivery

In its M-Pak Modular Packages, B&G presents a new advance in efficiency designing of air conditioning and refrigeration equipment. M-Pak units are built around five basic sizes of open-type compressors. Factory-stocked motors, condensers, evaporators and controls can be added to these basic units to make a comprehensive line of "packages" of from 7½ to 150 tons.

1. For example, motor compressors are produced by adding a B&G-built motor to a B&G basic compressor. **2.** Further addition of a B&G condenser and a standard base makes a condensing unit. **3.** Addition of evaporator and control panel results in a package liquid cooler.

All major components of M-Pak units are designed, manufactured, and guaranteed by B&G...*one* responsibility for the entire package. This, we believe, is an exclusive feature.

B&G also makes a complete line of refrigeration and air conditioning evaporators, condensers and centrifugal pumps—with many commonly used sizes available for immediate shipment from factory stock.

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COMPANY
Dept. GT-64, Morton Grove, Illinois

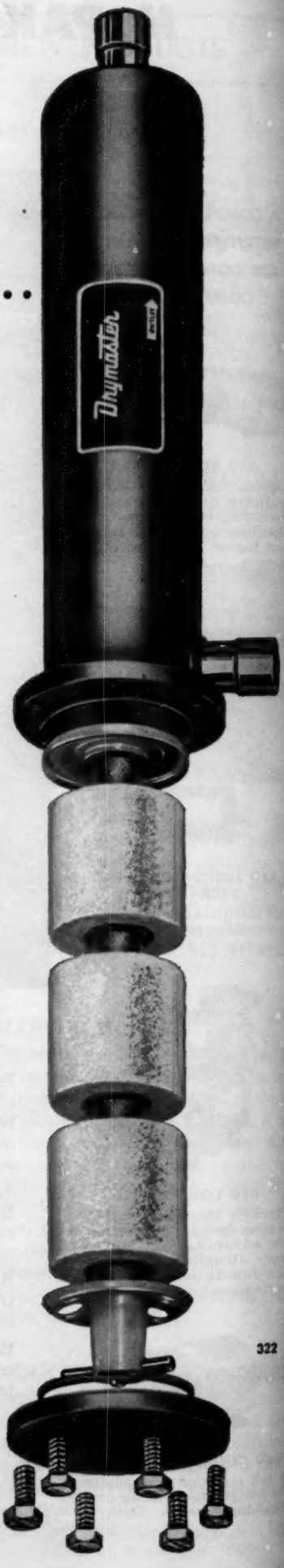
Canadian Licensee: S. A. Armstrong, Ltd. 1400 O'Connor Drive, Toronto 16, Ontario

UNSURPASSED FILTERING AND DRYING with ADDED Convenience and Economy... It's the New Refillable Type *Drymaster*® Filter-Drier!

The superb, balanced filtering and drying performance of the new refillable type DRYMASTER comes from HI-FI Filter Block Desiccant: These fully activated, pressure molded briquettes quickly remove all harmful moisture, acid, sludge and micron-size foreign particles from the refrigeration system. HI-FI Filter Block Dessicant is non-dusting, non-channeling and is virtually unequalled in moisture adsorbing capacity. Its combination surface and internal depth filtering ability is unsurpassed by any other comparably sized drier in the field today.

And the new angle-type, refillable DRYMASTER Filter-Drier gives added convenience and economy, too. Once installed, the body shell of the Drier is in for the life of the system; only the desiccant is replaced. DRYMASTER Refillable Filter-Driers are designed for 5 ton through 165 ton capacities with ½" O.D. through 2⅛" O.D. solder connections. End fittings are made of copper for easier soldering.

HI-FI Desiccant Briquettes, protected against shock by fibre glass padding, come packed in vacuum sealed cans ready to be installed quickly, easily, with no connections or joints to remake.



Write today for Catalog R-159 . . . contains complete information on all Streamline Refrigeration and Air-Conditioning products.

322

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PORT HURON 15, MICHIGAN

VAMPCO ALUMINUM PRODUCTS, LTD., STRATHROY, ONTARIO
Exclusive Canadian Representative for Mueller Brass Co.
Air Conditioning and Refrigeration Products

YOURS FOR THE ASKING...

over 80,000 prop blade combinations!

...and all at standard production run prices too. No expensive engineering or tooling costs to absorb. You name the application; electronics, heating, cooling, general appliances and Meier has a quality built propeller blade for it. Let Meier evaluate your product's impeller requirements in their own testing laboratory (no obligation, of course) and recommend the most efficient, economical prop blade you can use. Sample blades on request.

Meier

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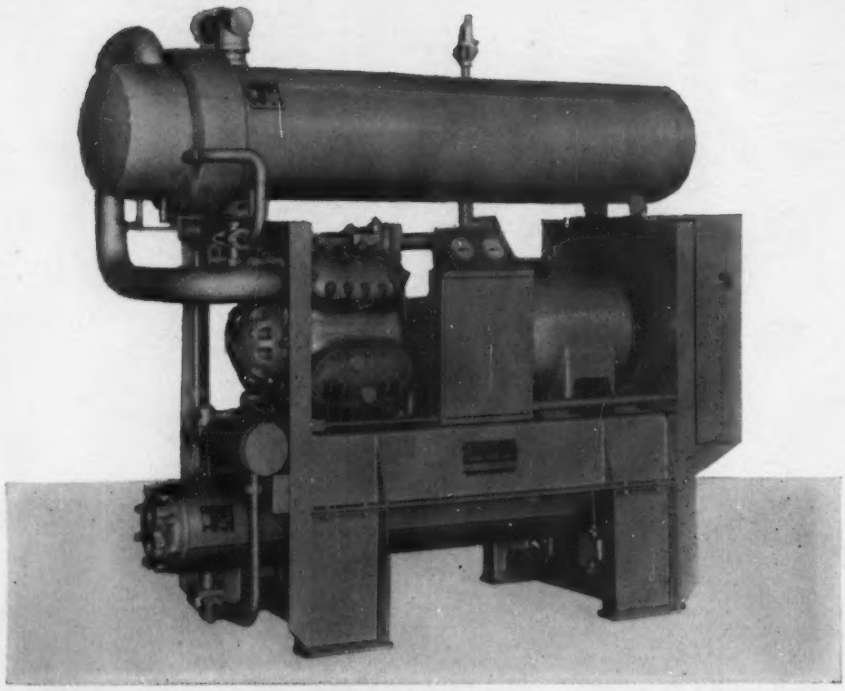
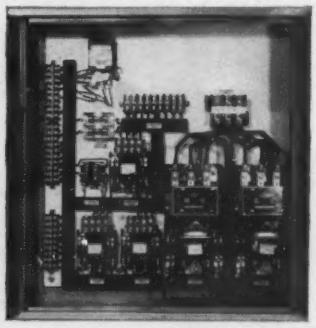


BELL & GOSSETT specifies Furnas Electric Controls

Rigid performance standards set by Bell & Gossett demand that only highest quality components be used in their equipment. For this reason Furnas Electric starters and contactors were chosen for B & G Package Liquid Chillers. These controls assure dependable operation and top-quality performance.



Furnas components in this B & G electrical control panel are mounted, wired and interlocked for trouble-free automatic operation.

In-between sizes, dual voltage coils, silver-cadmium oxide contacts and other popular Furnas features make Furnas controls well suited for your air conditioning and refrigeration requirements. Compact construction, front accessible wiring terminals and uniform mounting dimensions make them highly adaptable to simplified panel mounting.

Write today for full information—
1182 McKee St., Batavia Illinois

FURNAS

FURNAS
ELECTRIC COMPANY
BATAVIA, ILLINOIS

NEW PRODUCTS

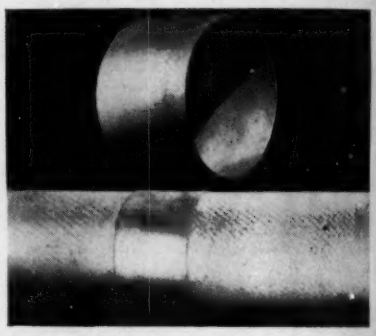
(*Continued from page* 104)

top. Standard vertical cabinet models are but six in. deep and 24 in. high. Low vertical cabinet models are nine in. deep and twelve in. high.

Ten models–standard vertical and horizontal, new high capacity vertical and horizontal and low vertical–are all available with and without cabinets. Eight nozzle arrangements, five unit sizes and three coils allow a wide range of selection flexibility.
Trane Company, LaCrosse, Wisc.

COUPLING

For use with Transite Air Duct, this new coupling is cited as assuring a watertight joint when used with asbestos-cement duct. A reinforced product

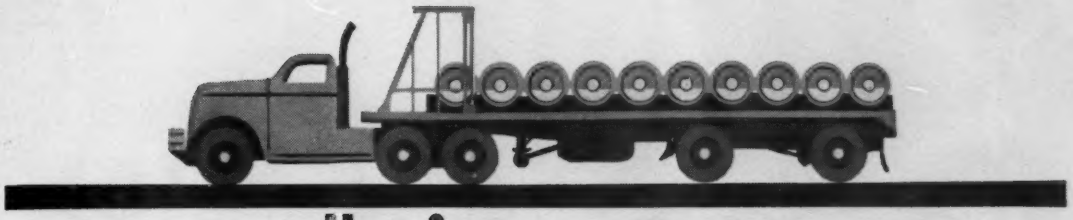


using inorganic asbestos and bonded with waterproofing ingredients, the unit is highly resistant to corrosion and is self-sealing with the Transite ducts and fittings it joins. Couplings are available in three, four, five, six, seven, eight, ten and twelve-in. sizes.
Johns-Manville Corporation, 22 E. 40th St., New York 16, N. Y.

PIPE JACKETING

To compensate effects of weathering on insulation covering outdoor runs of piping or duct-work, this company has introduced a laminated facing material consisting of aluminum foil, glass reinforcing strands and a new polyvinyl fluoride film. Bonding these sheets together is Pyro-Kure, a self-extinguishing adhesive.

Exposed to the atmosphere when the facing is applied, the polyvinyl film is resistant to flexural failure and is dimensionally stable at elevated temperatures. It is cited as retaining its characteristics over a wide temperature range, remaining flexible at 355 F and resistant to flexural fatigue at 0 F.
American Sisalkraft Company Div, St. Regis Paper Company, 55 Starkey Ave., Attleboro, Mass.



Ucon® refrigerants RACK TRUCK

—for speedy, on-the-spot delivery of one-ton cylinders.

Ucon® refrigerants COMPARTMENTED TANK WAGON

—for one-trip delivery of two grades of refrigerants, each compartment having 15,000-lb. capacity.

Ucon® refrigerants TANK CAR

—for economical rail delivery in 6, 8 and 10-thousand gallon capacities.

THREE WAYS TO ROLL UP SAVINGS!

...with the most complete delivery facilities in the refrigerants industry!

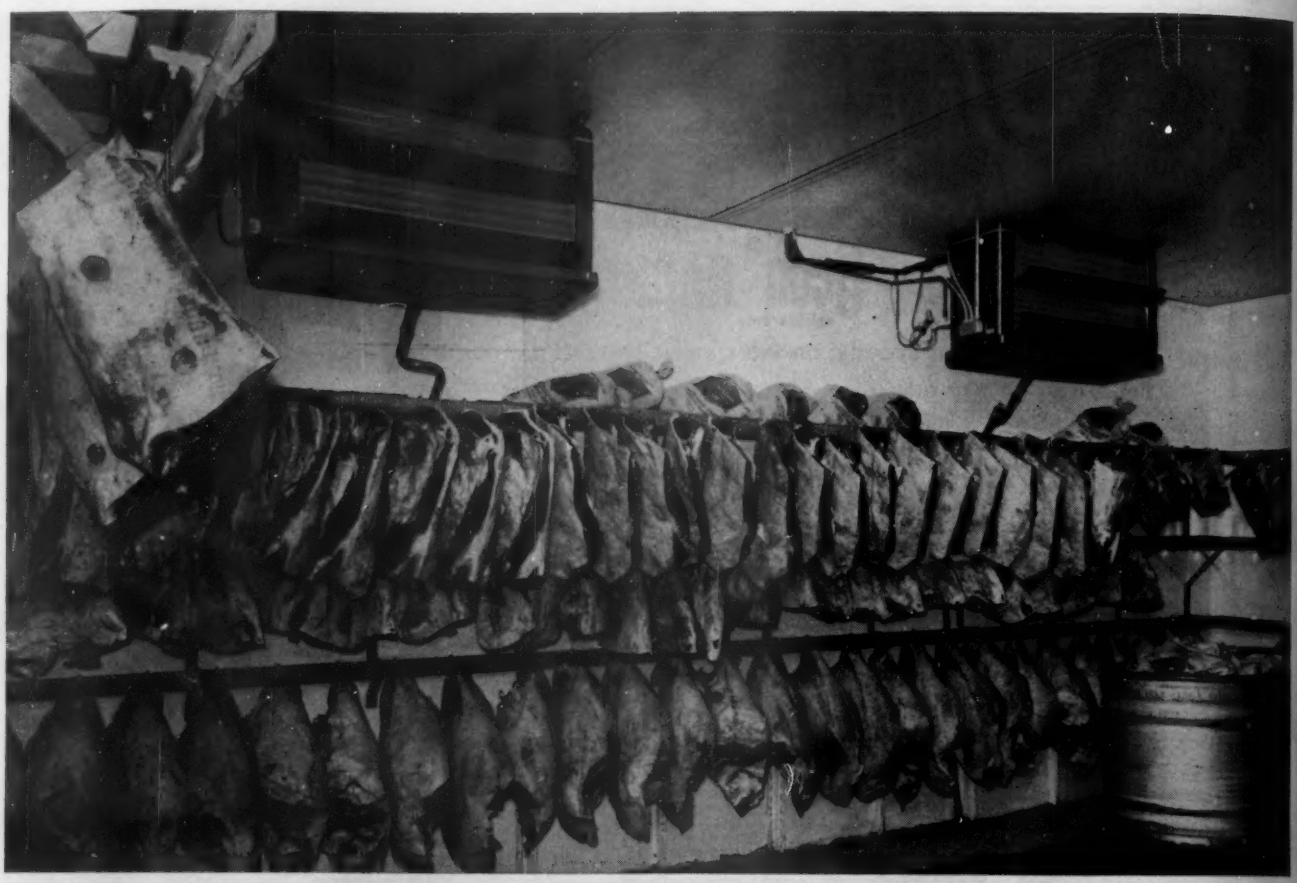
These are the vehicles that deliver Ucon® Refrigerants to you in bulk, reducing your costs through quantity discounts and lower freight rates. Countless man-hours in unloading, testing and handling are eliminated. Whatever your refrigerant requirements, one of these service vehicles can quickly, dependably and *economically* meet them!

Let Carbide help you enjoy the economy and convenience of bulk shipments!

- Free technical assistance available to help you establish bulk storage facilities.
- Complete design and specification data for the bulk installation best suited to your requirements.

We draw on many years of experience in handling more than 100 chemicals in bulk quantities! Your local Ucon Refrigerants representative has the whole, eye-opening story on these bulk delivery facilities and the *big savings* they offer. Call him today...and put this money-saving fleet to work for you! UNION CARBIDE CHEMICALS COMPANY, Division of Union Carbide Corporation, 270 Park Avenue, New York 17, New York.

UNION CARBIDE CHEMICALS



LARKIN MLT meat storage program saves money for retailers, makes money for refrigeration dealers

The retail meat industry is extremely enthusiastic about Larkin's new program for proper meat storage using the new LARKIN MLT System.

Astonishing graphic proof of the results obtained in two years of research and testing have sold this system to large chain and independent operations. Here's what the system does:

- CUTS MEAT SHRINKAGE
- REDUCES TRIMMING WASTES
- RETAINS BLOOM OF MEAT
- IMPROVES SHELF LIFE
- INCREASES PROFIT

The industry is buzzing. Stores are not only adopting the program for new construction but are also finding it profitable to convert existing systems.

LARKIN is ready to help you capitalize on this *new* source of revenue. Literature that tells the story, *sells your prospects*, is available to all LARKIN dealers.

Get in on the ground floor. See your wholesaler or write direct for Bulletin MLT-1061.

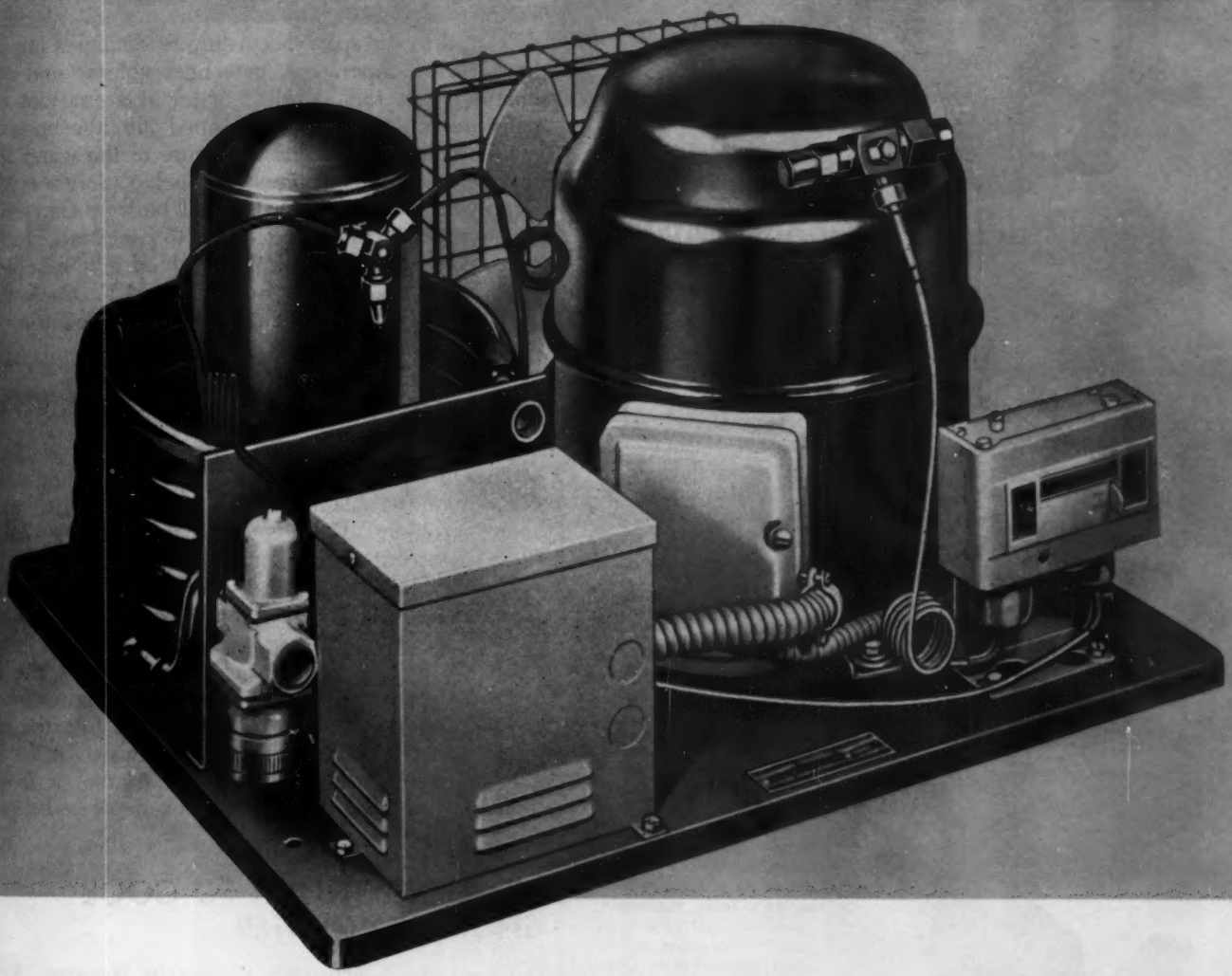
WATCHDOG OF THE NATION'S FOOD AND COMFORT ☆ LARKIN ☆

LARKIN COILS INC.

519 Memorial Drive, S.E. • P. O. Box 1699 • MUrray 8-3171

ATLANTA 1, GEORGIA

Tecumseh... Producer of Refrigeration Condensing Units... uses Wolverine Tube



WOLVERINE CAPILATOR®—the tiny, precision-made capillary tube for effective metering of liquids and gases – is an important contributor to the success of the hermetic condensing units manufactured by Tecumseh Products Company.

Tecumseh—one of the prominent producers of refrigeration compressors and condensing units uses Wolverine Capilator in its highside control system. Wolverine seamless copper tube is also used in suction and connecting lines.

So that it will meet the performance standards of Tecumseh and other American manufacturers, Capilator is always plug-drawn to insure smooth, mirror-bright inside diameters. Its ends are chamfered for unimpeded flow and, for maximum cleanliness, each length is washed in solvent, purged with "dry" air and has the ends paper wrapped. Capilator's manufacturing tolerances are rigidly controlled to meet customer's stated flow requirements.

If your company uses seamless copper, copper alloy or aluminum tube why not follow the lead set by Tecumseh and other leading American companies—specify Wolverine. Write, too, for your free copy of the Wolverine Tubemanship Booklet.

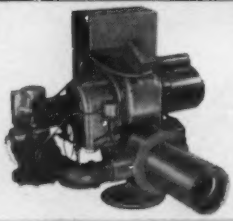
WOLVERINE TUBE
DIVISION OF
Calumet & Hecla, Inc.
DEPT. P, 17244 SOUTHFIELD RD., ALLEN PARK, MICH.
TUBEMANSHIP in Copper—Copper Alloys—Aluminum—Special Metals

9557

PLANTS IN DETROIT, MICHIGAN AND DECATUR, ALABAMA • SALES OFFICES IN PRINCIPAL CITIES.

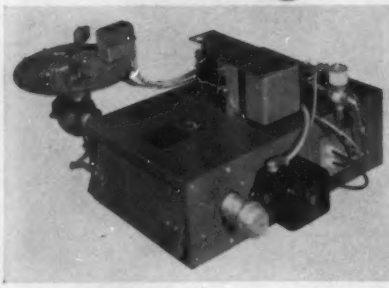
the Industry's most complete line of

CONVERSION BURNERS



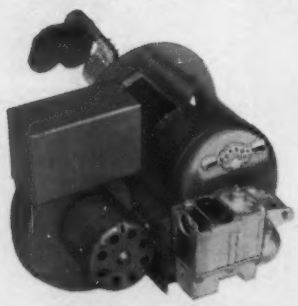
THE CYCLO JET

. . . is a complete fuel burning device and control system combining high efficiency and compact design for gas or dual fuel. Fully guaranteed and time tested.



THE MULTURI

. . . is a horizontal inshot or gun type gas burner — Ideal for commercial and industrial boilers as well as warm air furnaces and drying units. Factory assembled, with automatic valve installed — Input capacities to 9,450,000 BTU's.

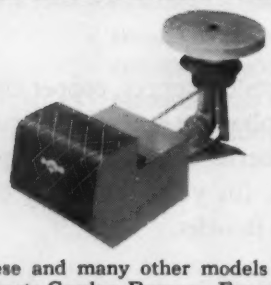


SERIES 450-P

. . . is a blower type gas conversion burner especially designed for difficult oil to gas conversion involving revertible flues and high resistence combustion chambers. Its telescoping mixer permits precise adjustment of mixes in combustion chamber. Capacities up to 500,000 BTU's.

SERIES 400R

Inshot type gas conversion burner. Specially developed to convert oil-designed furnace or boiler to a fully automatic gas fired unit at a fraction of the cost of replacing the entire unit. It has an exclusive, precision built, iris orifice affording perfect gas input adjustment.



SERIES 84

Gas conversion burner Up-Shot type with the Gordon Spreader-Flame. Designed to convert any coal or oil burning boiler or furnace up to the capacity of 900,000 BTU's into a fully automatic, effortless gas firing unit.

These and many other models make up the complete line of Roberts-Gordon Burners. For specifications, write . . .

Roberts-Gordon APPLIANCE CORPORATION

44-A3 CENTRAL AVE. BUFFALO 6, NEW YORK

In Canada: GRIMSBY, ONTARIO

Applications

LAGUARDIA TERMINAL TO HAVE METERED AIR CONDITIONING

Automatic sensing devices in each airline area at LaGuardia Airport will adjust the amount of cooling delivered to the space according to demands imposed by outside temperature, passenger volume and other fluctuating factors. Chilled water at a constant temperature of 44 F will be pumped into the units and meters will record the temperature of the water leaving the units to determine how much cooling was used. Supplying the chilled water will be four Carrier absorption refrigerating machines, to be installed at the east end of the new terminal, which will be able to cool 5240 gpm of water from 57 to 44 F. Each water chiller will have a cooling capacity of 682 ton.

AIR CONDITIONING THE ALAMO

Installation of air conditioning apparatus so that it would not present a contrast with the rough-hewn interior of the Alamo at San Antonio, Texas, was a major problem. High ceilings, the 100-deg span of Texas temperatures and stone walls several feet thick and unable to be insulated were other considerations. General Electric Weathertrons were installed in various areas to solve these problems.

Two four-ton units were concealed on the roof behind a parapet, with supply outlets extending through choir lofts below the roof and into the main room. Heavy iron window grills were utilized to disguise the outlets. Behind the Alamo, a five-ton unit was installed in a small stockade of cedar posts.

37,000,000 BTU/HR TO COOL SHOPPING CENTER

To cool the Sharpstown Center in Houston, Texas, the air conditioning system will provide more than 37,000,000 Btu/hr. The 3088-ton system, controlled by more than 100 thermostats, will change automatically to heating in winter and will maintain temperature and humidity over all of the 760,000-sq ft area. Combinations of automated controls allow the York system to determine when to idle, when to air condition only the mall and when to increase or decrease output for varying loads.

Design of the exhaust system prevents the individual stores, which open directly upon the mall, from becoming high pressure areas. Exhausted air from the stores is drawn into the mall through the open doors and special ducts by low pressure in the mall. Air conditioning units atop the Center then draw air from the mall, where part of it is re-cooled and pumped back into the Center. When the stores are closed, the 26 units serving the mall supply 2,340,000 Btu/hr for window-shoppers.